PUMPING STATION LOUISVILLE WATER COMPANY.

DUTY TRIAL

OF A

PUMPING ENGINE

FOR THE

LOUISVILLE WATER COMPANY,

LOUISVILLE, KY.

2d EDITION.

BUILT BY

I. P. MORRIS COMPANY,

OWNED AND CONDUCTED BY

The William Cramp & Sons Ship and Engine Building Co.,

PHILADELPHIA, PA.

1895.

Report of Experts on Engine No. 3.

To the President and Directors Louisville Water Company,
Louisville, Ky.:

Gentlemen—In accordance with your request, the contract trial of Pumping Engine No. 3 was made under our direction during the six days beginning at 3 P. M. April 25, 1894, and ending at 3 P. M. on May 1, 1894.

Description of Plant.

Pumping Engine No. 3 was designed by E. D. Leavitt, of Cambridgeport, Mass., and Charles Hermany, of Louisville, Ky., and built by the I. P. Morris Company, of Philadelphia, Pa. It is of the Leavitt type, having two vertical inverted cylinders, the piston rod of the high pressure cylinder being connected by a link to one end of a beam, and the low pressure similarly to the other end of the beam. The main shaft is at one end of the engine, and the connecting rod passes from a pin in the upper part of the beam to the crank pin.

The steam pistons have opposite motions in consequence of this arrangement, and the exhausts from the ends of the high pressure cylinder take place to the corresponding ends of the low pressure cylinder. The steam cylinders are jacketed all over with steam of boiler pressure, and the reheating receivers, of which there are two, use boiler pressure steam for the reheating medium.

Each steam cylinder is provided with four grid-iron valves operated by Leavitt cams. The point of cut-off in the high pressure cylinder is automatically determined by a centrifugal governor, while that on the low pressure cylinder is fixed.

The construction of this engine is of the most massive character, the weight being far greater than any other pumping engine of the same capacity.

The pumps are located directly under the engine, and are connected to the beam at such points that while the stroke of each steam piston is 10 feet, that of each pump plunger is 7 feet. The plungers work vertically and are of the differential type, so that each plunger draws water through the suction valves on the upward stroke, discharges it through the discharge valves on the downward stroke, and expels one-half of it into the main on each stroke. Each pump is therefore single acting on the suction side and double acting on the delivery.

The pump valves are small and have 5-16 inch lift. There are 143 suction valves and 124 discharge valves in each pump.

The engine is provided with a surface condenser. The air pump is vertical double acting, and is worked by a crank on the end of the beam shaft.

On account of the rise and fall of the Ohio River the bed plate of the engine is placed above the highest probable high water mark, while the bottoms of the pumps are sufficiently low to draft water at the lowest stages of the river. The distance from the bottoms of the pumps to the bottom of the bed plate is 61 feet.

The engine is provided with three boilers of the Belpaire locomotive type, built by the I. P. Morris Company, having a diameter of shell 82 inches inside of the smallest ring. The boilers are fed by a Worthington duplex pump, the exhaust of which passes into an open heater. The feed water is that from the condensed steam of the main engine, there being provision for making up with cold water any waste that may occur.

By the terms of the contract between the Louisville Water Company and the I. P. Morris Company, it was specified that—

"The trial shall be commenced by taking the engine in its current working condition, while running and pumping the ordinary daily supply of water into the reservoir; the boilers, however, shall be clean, and all appurtenances of the machine in proper working order. While in this state of service the engine shall be stopped at about 10 o'clock in the morning, the fires hauled, the ash-pit, furnaces, and boiler-room cleared of all fuel and combustible, and all steam blown off from the boilers, and the water therein brought to the normal level. The fires shall then be started fresh, and all the fuel used thenceforward shall be weighed and charged to the engine; the wood used to start the fires being rated at one-half its weight as coal.

"As soon as sufficient steam has been generated to turn the engine over against its regular water load, it shall be started and run, uninterruptedly, for one hundred and forty-four hours, and stopped when the steam pressure in the boilers shall have been worked down to a point at which the engine will no longer revolve against its load. All the coal and other fuel consumed during these one hundred and forty-four hours shall be charged to the engine in its merchantable form and condition, and no picking out of inferior coal, or deduction for ash, clinker, cinder, or incombustible of any kind shall be permitted or made.

"The coal used during seventy-two hours of the trial shall be such as is currently used at the station, viz.: The best quality, second pool, Pittsburg bituminous coal, and during the remaining seventy-two hours Cumberland bituminous coal shall be used. The head against the pumps, from which the duty is to be computed, shall be the height between the water-levels in the pump-well and stand-pipe, both taken at a mean height, to which is to be added the head due to the friction encountered by the water in its passage from the well through the valves, pumps and pipes, to the stand pipe.

"At the end of each twenty-four hours, and before the expiration of the succeeding twenty-four hours, the account of the engine's performance shall be made up and reported, in writing, to both parties in this contract; and during each such period of twenty-four hours into which the time of the trial is to be subdivided, the pumps shall deliver into the reservoir not less than sixteen million (16,000,000) U. S. gallons of water, the volume of which is to be determined by gauging over a weir at the reservoir; and the engine and pump shall make a duty of not less than 85,000,000 pounds of water raised one foot high per hundred pounds of Pittsburg coal consumed, and 110,000,000 with Cumberland coal.

"The formula for computing the duty shall be as follows:

$$D = \frac{G \times 8.34 \times (H + h)}{W \times \frac{1}{100}}$$ in which

"D=The duty.

"G=The number of gallons of water delivered into the reservoir in twenty-four hours.

"8.34=The weight per gallon of water.

"H=Mean difference between the water levels in the well and stand-pipe.

"h=The head due to the friction encountered by the water in passing from the well to the stand-pipe.

"W=The total weight of coal in pounds consumed per twenty-four hours."

The above requirements were carried out as far as practicable. By mutual agreement the steam was not blown off from the boilers, and Pocahontas coal was used instead of Cumberland, as it was more easily obtainable.

The method of conducting the trial was as follows:

The engine was run a number of hours up to the trial speed and then stopped. The fires were drawn, the boiler tubes blown out, and

new fires started. As soon as steam began to rise the engine was started. Steam pressure was increased to the working pressure of 140 pounds, and the engine speeded up to about 18.6 revolutions per minute as rapidly as possible, and run 144.17 hours continuously.

Boilers Nos. 2 and 3 were used.

The coal was weighed in barrows as needed, separate accounts for each boiler being kept.

Samples of the coal were taken daily, and the amount of moisture ascertained by drying for 24 hours on top of the boiler flue.

The main feed water was weighed on accurate scales, and the jacket and reheater condensations were returned to the boiler by gravity through a Worthington meter. This meter, although dealing with water of 328° F. temperature, worked with perfect uniformity throughout the run. At the end of the trial it was calibrated under the working conditions by a run of three hours.

The feed water was weighed before passing through the feed-water heater, and the amount of water added to the feed by the condensation of the feed pump exhaust was computed from the rise in temperature of the feed caused thereby, and from the work done by the pump. The heat radiated by the pump and its steam pipe was necessarily ignored.

Indicator diagrams were taken every hour throughout the trial, and readings of the engine counter, steam pressures at boiler and engine, vacuum, barometer, force main guage, height of water in well, temperatures of air, steam, feed water, and escaping gases from flue were taken every thirty minutes.

All of these readings, as well as the weights of the coal, were taken simultaneously by two observers, representing both parties to the contract.

The scales used for weighing the coal and feed water were adjusted and tested on the day preceding the trial.

All thermometers and gauges were carefully tested. The head on the pumps was determined by a mercurial column connected with the pump chamber just above the delivery valves. To the reading of this gauge was added the distance from the zero of the mercury column to the surface of the water in the pump well. While the friction head through a portion of the pump and the whole of the delivery main is thus credited to the engine, that of the water through the lower part of the pump and pump valves is not.

A calorimeter was used on the steam pipe near the engine, and a separator removed the water of condensation from the steam pipe.

This condensation was drained into the air pump discharge, and thus raised the temperature of the feed water. The rate of discharge of the separator was ascertained by a special run made after, and under the conditions of the trial.

Indicator diagrams were occasionally taken from each main pump.

Throughout the trial the discharge over the weir was gauged continuously by competent observers.

In every part of the plant precaution was amply taken to either avoid or accurately account for all leakage toward or away from the plant.

It should be noted that the engine is situated so far from the boiler house that the length of the main steam pipe is 180 feet.

The steam used by the feed pump and the water formed by its condensation were considered as passing from and to the boiler at a uniform rate, and in a cycle independent of that of the steam used by the main engine. In other words, this water was deducted from the total feed in determining that used by the main engine.

The Pittsburg coal was used during the first three days of the trial, and in changing from this to Pocahontas coal the effort was made to begin firing the Pocahontas coal at such a time that at the specified hour of change there was one-half of each kind of coal on the grates. The effect of this was to give too low an average duty with the Pittsburg coal and too high with the Pocahontas, for the Pittsburg coal raised the steam from the original to the working pressure, and at the end the Pocahontas coal was allowed to burn out until the working pressure had fallen to the original. The wording of the contract, however, allowed no other method of procedure. It is probable that the results of the second day with each coal represent the performance of the coals, and furnish the best means of comparison.

After the trial the indicator spring scales were determined by the U. S. Navy Department, at the New York Navy Yard.

The following tables give the principal dimensions of the engine and boilers, with the results of the trial in detail:

DIMENSIONS OF THE ENGINE.

Type—Leavitt compound vertical, inverted, beam, fly-wheel.

1. Diameter of high pressure cylinder, hot, ... 27.21 in.
2. Diameter of low pressure cylinder, hot, ... 54.13 in.
3. Diameter of fly-wheel, ... 36 ft.
4. Diameter of H. P. piston rod, ... 5½ in.

5. Diameter of L. P. piston rod,..6 in.
6. Stroke of each piston,..10 ft.
7. Mean clearance H. P. cylinder,........................1.585 per cent.
8. Mean clearance L. P. cylinder,........................1.530 per cent.
9. Diameter of each differential plunger,......34 in. and 24 1 16 in.
10. Stroke of each differential plunger,....................................7 ft.
11. Ratio of steam piston areas,..4 to 1
12. Volume displaced by one stroke of one plunger,......330.15 gal.
13. Volume displaced by plungers during one revolution of engine,..660.30 gal.
14. Diameter of each discharge pipe,....................................24 in.

DIMENSIONS OF THE BOILERS.

15. Type—Belpaire Locomotive, Double Furnace.
16. Number in use,...2
17. Smallest inside diameter of shell,...................................82 in.
18. Inside length of fire box,.....................................8 ft. 1½ in.
19. Inside width of fire box, twice 3 ft. 9 in ,.................7 ft. 6 in.
20. Length of combustion chamber,.......................................7 ft.
21. Length of tubes,..16 ft.
22. Diameter of tubes outside,...3 in.
23. Number of tubes, each boiler,..159
24 Length of grate, each boiler,................................7 ft. 1 in.
25. Width of grate during trial bricked up in each boiler to 5 ft. 3 in.
26. Heating surface, each boiler,.............................2,240 sq. ft.
27. Grate surface during trial, each boiler,.................37,185 sq. ft.
28. Ratio of heating to grate surface during trial,............60.24 to 1

ENGINE TRIAL.

29. Duration of trial, ... 144 hrs. 10 min.
30. Duration of trial with Pittsburg coal, ... 72 hrs.
31. Duration of trial with Pocahontas coal, ... 72 hrs. 10 min.
32. Total number of revolutions, ... 160,666.5
33. Average number of revolutions per minute, ... 18.574
34. Average steam piston speed, feet per min .... 371.48 ft.
35. Average water plunger speed, feet per min. .... 160.04 ft.

AVERAGE TEMPERATURES.

36. Of engine-room, ... 60° to 86° F.
37. Of external air, ... 48° to 86° F.
38. Of main feed water at weighing tank, ... 81.2° F.
39. Of main feed water on entering boilers, ... 108° F.
40. Of jacket and reheater drain at boilers, ... 328.3° F.
41. Of mixture of feed waters at boilers, ... 143.3° F.
42. Of water in pump well, ... 58.7° F.

AVERAGE PRESSURES.

43. Of atmosphere by barometer, ... 14.60 lbs.
44. Of steam at boilers by gauge, ... 140.00 lbs.
45. Of steam at boilers absolute, ... 154.60 lbs.
46. Of steam at engine by gauge, ... 137.00 lbs.
47. Of steam at engine absolute, ... 151.60 lbs.
48. Absolute initial pressure H. P. cylinder, ... 145.75 lbs.
49. Absolute terminal pressure L. P. cylinder, ... 7.32 lbs.
50. Absolute back pressure L. P. cylinder, ... 0.95 lbs.
51. Vacuum by gauge, ... 27.75 in.
52. Ratio of expansion, ... 20.40
53. Water pressure by mercury column, ... 62.50 lbs.
54. Height of mercury zero above water in pump well, ... 49.04 ft.
55. Total water pressure, ... 83.74 lbs.
56. Equivalent head, ... 193.15 ft.

STEAM USED BY ENGINE AND FEED PUMP.

57. Weighed feed water, ... 968,128 lbs.
58. *Feed pump steam, condensed, ... 25,390 lbs.

*This was condensed by the weighed feed water in an open heater and then pumped into the boiler with the weighed feed.

59. Total water pumped into boilers,. ....991,518 lbs.
60. Total water returned to boilers from jackets and reheaters,..189,795 lbs
61. Total steam used by calorimeter,...............................727 lbs.
62. Total water drained from separator........................23,428 lbs.
63. Total moist steam used by engine and feed pump...1,157,158 lbs.
64. Percentage of moisture in steam after leaving separator,..0.55 per cent.
65. Total dry steam used by engine and feed pump,....1,150,792 lbs.

STEAM USED BY ENGINE.

66. Total moist steam used by engine only,...............1,133,768 lbs.
67. Total dry steam used by engine only,.................1,127,533 lbs.
68. Total moist steam passing through cylinders,...........943,973 lbs.
69. Total moist steam passing through jackets and reheaters,..189.795 lbs.
70. Percentage of moist steam used by jackets and reheaters,...16.74 per cent.
71. Moist steam used per hour per I. H. P.,.................12.223 lbs.
72. Dry steam used per hour per I. H. P......................12.156 lbs.
73. Dry steam passing through cylinders per hour per I. H. P.,..10.120 lbs.
74. Moist steam used per hour per pump H. P.,.............13.123 lbs.
75. Dry steam used per hour per pump H. P.,...............13.050 lbs.
76. Prevailing point of cut-off, H. P. cylinder,.........20.20 per cent.
77. Prevailing point of cut-off, L. P. cylinder,.........42 10 per cent.
78. Ratio of expansion, by volumes,................. ....20.40
79. Steam accounted for by indicator at H. P. cut-off, 7.75 lbs.,=..76.58 per cent.*
80. Steam accounted for by indicator at H P. release, 9.166 lbs.,=..90.57 per cent.*
81. Steam accounted for by indicator at L. P. cut-off, 10.008 lbs.,=..99.60 per cent.*
82. Steam accounted for by indicator at L. P. release, 9.725 lbs.,=.. .....96.09 per cent.*

BRITISH THERMAL UNITS USED BY ENGINE AND FEED PUMP.

83. Heat of vaporization of steam, 154.6 lbs. absolute, 859.4 B. T. U.
84. Heat of liquid of steam, 154.6 lbs. absolute,......332.5 B. T. U.
85. Heat of liquid, jacket and reheater drain, 328.3° 298.7 B. T. U.

*Percentage of 10.12 lbs.

86. Heat of liquid, main feed, 108.°,........................76. B. T. U.
87. Per lb. of moist steam used by cylinders of main engine
and feed pump, 859.4 x .9945 + 332.5—76=1111.2 B. T. U.
88. Per lb. of moist steam used by jackets and re-
heaters,.........859.4 x 0.9945 + 332.5—298.7=888.5 B. T. U.
89. Total used by engine and feed pump during trial, 1,243,566,000
90. Total used by engine and feed pump in 24 hours,...207,016,600

BRITISH THERMAL UNITS SUPPLIED BY BOILERS.

91. Heat of vaporization, steam 154.6 lbs. absolute, 859.4 B. T. U.
92. Heat of liquid, steam 154.6 lbs.,........................332.5 B. T. U.
93. Heat of liquid feed. 143.3°,..............................111.5 B. T. U.
94. Per lb. of moist steam supplied by boil-
ers,.........859.4 x 0.9945 + 332.5—111.5 = 1,075.7 B. T. U.
95. Total supplied by boilers,..1,181,313 x 1075.7 = 1,270,738,600
96. Total supplied by boilers in 24 hours.........................211,540,000
97. Total supplied per lb. Pittsburg coal,..................................9,198
98. Total supplied per lb. Pocohontas coal,...............................10,350

BRITISH THERMAL UNITS USED BY ENGINE.

99. Per lb. of moist steam used by the cylinders,...1134.5 B. T. U.
100. Per lb. of moist steam used in jackets and re-
heaters,...880.8 B. T. U.
101. Used by engine during trial,..................1,238,108,959 B. T. U.
102. Used by engine in 24 hours,......................206,111,428 B. T. U.
103. Used by engine per minute per I. H. P..............222.46 B. T. U.
104. Thermodynamic efficiency of engine,..................19.07 per cent.

AVERAGE POWERS, ETC.

105. Total number revolutions in 144 hrs. 10 min.,.............160,666.5
106. Average number revolutions per minute,..........................18.574
107. Average piston speed, feet per minute,............................371.48
108. Average plunger speed, feet per minute............................260.04
109. Average mean effective pressure in H. P. cylinder,......43.53 lbs.
110. Average mean effective pressure in L. P. cylinder,...14.155 lbs.
111. Horse power developed by H. P. cylinder,...........279.00 H. P.
112. Horse power developed by L. P. cylinder,...........364.40 H. P.
113. Horse power developed by both cylinders,..........643.40 H. P.
114. Percentage of power developed by H. P. cylinder,............43.36
115. Percentage of power developed by L. P. cylinder,............56.64

116. Horse power of plungers,..................................599 10 H. P.
117. Friction horse power,...44 30 H. P.
118. Efficiency of mechanism,...................................93.12 per cent.
119. Friction of mechanism.......................................6.88 per cent.

BOILER TRIAL.

		Kind of Coal. Pittsburg.	Pocahontas.
120.	Number of boilers in use,....................	2	2
121.	Duration, hours,..................................	72.08	72.167
	Average Pressures.		
122.	Steam pressure at boilers by gauge, lbs.,...	140	140
123.	Atmospheric pressure by barometer, lbs.,..	14.59	14.62
124.	Absolute steam pressure, lbs.,..............	154.59	154.62
125.	Force of draft in inches of water, in.,......	0.5	0.5
	Average Temperatures.		
2 6.	Of escaping gases from boilers, deg.,........	438.5	457.5
127.	Of feed water before entering heater, deg.,	79.0	83.33
128.	Of feed water on entering boilers, deg.,...	142.2	144.5
	Fuel.		
129.	Moist coal consumed, lbs.,.....................	67,917	63,591
130.	*Wood consumed at 50 per cent. weight lbs.,	772	25
131.	Moisture in coal, per cent.,..................	0.7	2.6
132.	Dry coal consumed with wood equivalent lbs.,	67,995	61,692
133.	Total refuse, dry lbs.,..........................	2,025	2,849
134.	Total refuse, dry, per cent.,..................	2.98	4.62
135.	Total combustible, lbs.,........................	65,970	58,843
136.	Dry coal consumed per hour, lbs.,.........	943	855
137.	Combustible consumed per hour, lbs.,.....	915	815
138.	Calorific value of one pound of coal by analysis, B. T. U.,................................	13,226	14,924
	Quality of Steam.		
139.	Moisture,..	.0055	0.0055
	Water.		
140.	Pumped into boilers and apparently evaporated, lbs.,..	587,740	593,573

*Specified by contract.

141. Actually evaporated corrected for quality of steam, lbs.,	584,507	590,308
142. Factor of evaporation,	1.120	1.118
143. Equivalent water evaporated into dry steam from and at 212°, lbs.,	654,648	659,964
144. Equivalent water evaporated into dry steam from and at 212°, per hour, lbs.,	9,082	9,145

Economic Performance.

145. Water actually evaporated per pound of dry coal, lbs.,	8.60	9.57
146. Equivalent water evaporated per pound of dry coal from and at 212°, lbs.,	9.63	10.70
147. Equivalent total heat derived from a pound of dry coal, B. T. U.,	9,250	10,294
148. Water actually evaporated per lb. of combustible, lbs ,	8.86	10 03
149 Equivalent water evaporated per lb. of combustible from and at 212°, lbs.,	9.92	11.22
150. Efficiency of boilers, per cent.,	70.0	69.0

Rate of Combustion.

151. Dry coal burned per square foot of grate per hour, lbs.,	12.70	11.50

Rate of Evaporation.

152. Water evaporated from and at 212° per square foot of heating surface per hour, lbs.,	2.02	2.04

Commercial Horse Power.

153. On basis of 30 lbs. of water per hour evaporated from 100° into steam of 70 lbs. gauge pressure (=34½ lbs. from and at 212°), H. P.,	263.2	265.1

Coal.

154. Used per indicated horse power per hour, lbs.	1.47	1.33
155. Used per pump horse power per hour, lbs.,	1.58	1.43

Results of Coal Analysis.

		Pittsburg.	Pocahontas
156.	Moisture,	1.78 per cent.	0.73 per cent.
157.	Carbon,	75.65 per cent.	83.31 per cent.
158.	Hydrogen,	5.00 per cent.	5.15 per cent.
159.	Oxygen,	9.73 per cent.	4.65 per cent.
160.	Nitrogen,	1.50 per cent.	1.25 per cent.
161.	Volatile sulphur,	0.64 per cent.	0.76 per cent.
162.	Ash	5.70 per cent.	4.15 per cent.
		100.00 per cent.	100.00 per cent.

Moisture in Coal.

		Weight before drying, oz	Weight after drying oz.	Moisture, oz.	Percentage of Moisture.
163.	First day	492	488	4.0	0.7
164.	Second day,	182	181	1.0	0.6
165.	Third day,	178.5	177	1.5	0.8
166.	Fourth day,	188.5	184.25	4.25	2.25
167.	Fifth day,	214	208	6.00	2.80
168.	Sixth day,	346.75	337.25	9.50	2.74
169.	Average moisture in Pittsburg coal,				0.70
170.	Average moisture in Pocahontas coal,				2.60

DUTIES AND CAPACITY.

The contract requires a statement of the duty performed during each twenty-four hours by 100 lbs. of dry coal consumed for water delivered into the reservoir, as determined by weir measurement. For this reason the duties on this basis, as well as on others, will be stated for each day. The duties per 1,000,000 heat units consumed by the engine given in the tables are computed with temperatures of rejection of the steam cylinders taken at the main feed as it enters the boilers, and of the jackets and reheater drains taken at the boilers. The steam used in this computation includes that used by the feed pump.

In the general computation for the efficiency of the engine alone, (see Table, page 17), the temperature of rejection of the cylinders was taken at the air pump discharge, and of the jackets and reheaters at the collecting tank at the engine. The steam used by the feed pump was excluded.

DAILY DUTIES PER 100 LBS. OF DRY COAL.

DAY.	Daily Capacity Weir Measurement.	Daily Exc. over 16,000,000 Gallons.	Daily Capacity Plunger Displacement.	Slip Per Cent.	Duty by Weir Measurement.	Excess of Duty over Guarantee Per Cent.	Duty by Plunger Displacement.	Kind of Coal.
	GALLONS.	PER CENT.	GALLONS.		FT. LBS.		FT. LBS.	
First	16,374,400	2.34	17,574,900	6.83	109,187,000	28.46	117,192,000	Pittsburg.
Second	16,482,000	3.01	17,670,300	6.72	121,416,000	42.84	130,170,000	"
Third..................	16,467,600	2.92	17,641,900	6.66	121,211,000	42.60	129 854,000	"
Av. Pittsburg Coal...	....	......	....	......	116,993,000*	37.64*	125,444,000*	"
Fourth.................	16,511,500	3.20	17,696,400	6.70	123,356,000	12.14	132,208,000	Pocahontas.
Fifth	16,488,000	3.05	17,689,700	6.80	131,186,000	19.26	140,747,000	"
Sixth..................	16,613,000	3.12	17,814,900	6.75	134,915,000	22.65	144,676,000	"
Av. Pocahontas coal	....	......	....	......	129,653,000*	17.87*	139,031,000*	"
Average...............	16,489,420	3.06	17,681,350	6.74	....	....	....	....

*By independent computation.

DAILY DUTIES BASED UPON PLUNGER DISPLACEMENT.

DAY.	100 Lbs. Dry Pittsburg Coal.	100 Lbs. Dry Pocahontas Coal.	100 Lbs. Pittsburg Combustible.	100 Lbs. Pocahontas Combustible.	1,000,000 Heat Units.	1,000 Lbs. Moist Steam.	1,000 Lbs. Dry Steam.
	FT. LBS.	FT. LBS.	FT. LBS.	FT. LBS.	FT. LBS.	FT. LBS.	FT. LBS.
First	117,192,000	....	120,504,000	....	138,008,000	148,285,000	149,104,000
Second	130,170,000	....	133,895,000	....	137,525,000	147,892,000	148,710,000
Third	129,854,000	....	134,464,000	....	137,948,000	148,195,000	149,014,000
Fourth	....	132,208,000	....	138,516,000	137,387,000	147,667,000	148,483,000
Fifth	....	140,747,000	....	147,015,000	138,274,000	148,538,000	149,360,000
Sixth	....	144,676,000	....	152,370,000	136,260,000	146,459,000	147,269,000
Average	125,444,000*	139,031,000*	129,295 000*	145,762,000*	137,565,000*	147,837,000*	148,655,000*

*By independent computation.

NOTE.—The duties given in the last three columns include the steam used by the feed pump.

GENERAL COMPUTATION FOR EFFICIENCY OF ENGINE ALONE.

Duration of trial,...144 hrs. 10 min.
Total number of revolutions,.......................................160,666.5
Revolutions per minute,...18.574
Average I. H. P.,...643.40
Average pump horse power,..599.10
Friction H. P.,..44.30
Friction H. P. per cent.,..6.88
Efficiency of mechanism, per cent.,.......................................93.12
Total dry steam used by engine,.............................1,127,533 lbs.
Total moist steam used by engine,..........................1,133,768 lbs.
Total moist steam used by cylinders,..........................943,973 lbs.
Total moist steam used by jackets and reheaters,.............189,795 lbs.
Total moist steam used by jackets and reheaters, per cent.,........16.74
Dry steam used per I. H. P. per hour,..............................12.156 lbs.
Moist steam used per I. H. P. per hour,............................12.223 lbs.
Dry steam used per I. H. P. per hour by cylinders,...........10.120 lbs.
Mean absolute steam pressure at engine,..........................151.60 lbs.
Mean temperature of rejection of engine (air pump discharge), 84.2°
Mean temperature of rejection of jackets and reheaters,...........335.3°
Heat of liquid of air pump discharge,........................52.27 B. T. U.
Heat of liquid of jacket and reheater drain ....306.0 B. T. U.
Heat of vaporization of steam supply,......................860.6 B. T. U.
Heat of liquid of steam supply,..............................330.9 B. T. U.
*Dry steam in mixture used by engine,....................................0.9945
B. T. U. per pound of moist steam used by cylinders,..........0.9945 x 860.6 + 330.90—52.27 = 1134.50 B. T. U.
B. T. U. per pound of moist steam used by jackets and reheaters,............0.9945 x 860.6 + 330.9—360 = 880.8 B. T. U.
B. T. U. passing through cylinders in $144\frac{1}{6}$ hrs., 1,070,937,531 B. T. U.
B. T. U. passing through jackets and reheaters in $144\frac{1}{6}$ hrs.,...167,171,428 B. T. U.
B T. U. passing through engine,.................1,238,108,959 B. T. U.
B. T. U. used per I. H. P. per minute (moist steam), 222.46 B. T. U.

Thermodynamic efficiency of engine $\frac{33000}{222.46 \times 778} = 19.07$ per cent.

Plunger work performed in $144\frac{1}{6}$ hrs.,...........171,015,314,960 ft. lbs.

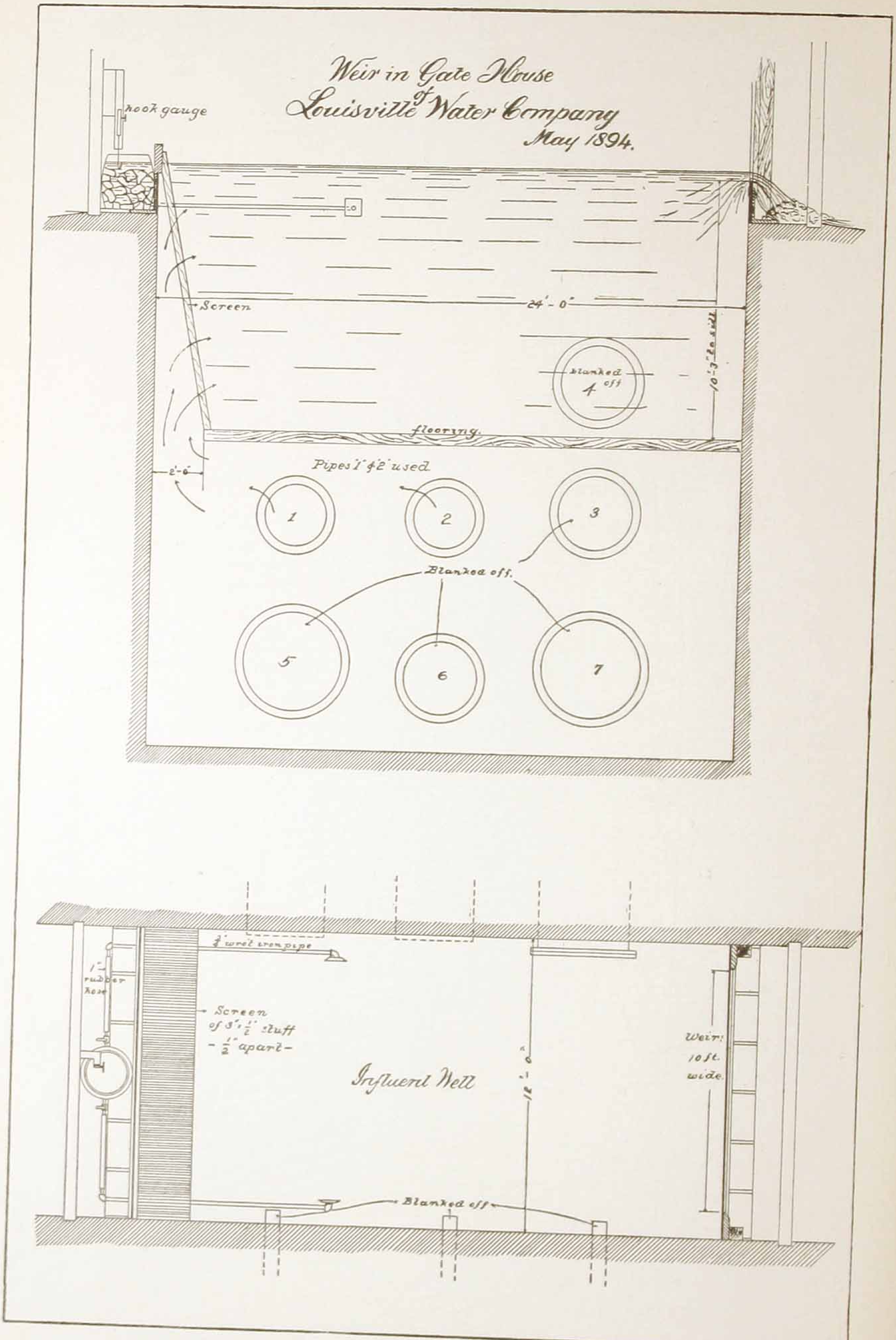
* Determined by calorimeter near engine.

*Mean duty per 1,000,000 B. T. U. used by engine alone,..138,126,000 ft. lbs.

*Mean duty per 1,000 lbs. moist steam used by engine alone,..150,838,000 ft. lbs.

*Mean duty per 1,000 lbs. dry steam used by engine alone,..151,672,000 ft. lbs.

*Based upon plunger work.


Weir in Gate House of Louisville Water Company May 1894.
hook gauge
Screen
24'-0"
10'-3" to sill
blanked off
4
flooring.
2'-6"
Pipes 1" & 2" used
1
2
3
Blanked off.
5
6
7
3/4" wrot iron pipe
1" rubber hose
Screen of 3" 1/2" stuff - 1/2" apart -
Influent Well
12'-0"
Weir: 10 ft. wide.
Blanked off

THE WEIR AND WEIR MEASUREMENTS.

The water pumped by the engine was forced through two lines of 30-inch Pipe to the distributing reservoir. At the reservoir the water was delivered into a masonry chamber 24 feet long, 12 feet wide, and 23 feet deep. At one end of this chamber there was a permanent weir having an edge of brass .50 of an inch in width, finished perfectly level, with a sharp edge on the upstream side. The weir extended entirely across the chamber, but during the trial the width of the weir was contracted to 10 feet.

About 7 feet above the bottom of the chamber a tight floor was laid, covering the entire bottom with the exception of an opening 2 x 12 feet, next the wall of the chamber, at the end opposite the weir.

The water entered the chamber under the temporary floor, passed up through the 2 x 12 feet opening at the end, and then passed through a rack 10 feet high and 12 feet wide, made of ½-inch strips of wood, with ½-inch spaces between, extending from the floor to the top of the chamber.

For the purpose of obtaining the head on the weir, brass plates 8 inches square, $\frac{3}{16}$ inch thick, having sharp edges, were placed at both sides of the channel 18 feet back from the weir and 6 inches below the crest. The faces of these plates were smooth, plane surfaces, placed parallel with the sides of the chamber and the direction of the current.

A hole ¾ of an inch in diameter, in the centre of these plates was connected by ¾-inch pipes, with a cask in which the water level was observed by means of a hook gauge. Very careful measurements were made to establish the zero of the hook gauge at the level of the crest of the weir, and also to test the height of the weir at different points.

During the whole trial observations of the depth of water flowing over the weir were taken every 10 minutes, and during the first and last hours they were taken every minute. The water in the channel of approach and while flowing over the weir was carefully protected from the effect of the wind.

The quantity delivered has been calculated by the well known Francis formula, as the conditions complied more closely with those

obtaining during his experiments than with those of the later experiment of Fteley and Stearns.

Formula:

$$Q=3.33\left[L-0.1n[(H+h)^{\frac{3}{2}}-h^{\frac{3}{2}}]^{\frac{2}{3}}\right]\left[(H+h)^{\frac{3}{2}}-h^{\frac{3}{2}}\right]$$

Q = Discharge in cubic feet per second.

H = The observed depth of water on the weir.

h = .0006 the height due to the velocity of approach.

n = 2 the number of end contractions.

L = 10 feet the length of the weir.

The head was very uniform, seldom varying more than .003 of a foot during an hour.

The average observed depths of water and resulting quantities delivered over the weir were as follows:

	H	Gals. delivered over weir.
3:00 P. M. to 3:10 P. M. April 25		55,200
3:10 P. M. to 3:20 P. M. April 25	0.786	102,030
3:20 P. M. to 3:30 P. M. April 25	0 804	106,020
3:30 P. M. to 4:00 P. M. April 25	0.8301	339,150
4:00 P. M. April 25 to 3:00 P. M. April 26	0.8452	15,772,000
3:00 P. M. April 26 to 3:00 P. M. April 27	0.84606	16,482,000
3:00 P. M. April 27 to 3:00 P. M. April 28	0.84555	16,467,600
3:00 P. M. April 28 to 3:00 P. M. April 29	0.84716	16,511,500
3:00 P. M. April 29 to 3:00 P. M. April 30	0.84624	16,488,000
3:00 P. M. April 30 to 3:00 P. M. May 1	0.84666	16,499,550
3:00 P. M. May 1 to 3:10 P. M. May 1	0.8416	113,450
Total for 144 hrs. 10 min.		98,936,500

The quantity delivered into the reservoir during the six days, as determined by weir measurement, was 6.74 per cent. less than the calculated displacement of the pump plungers.

As this percentage of difference or slip was larger than is generally obtained, tests were made of the leakage from the force main and through the pump valves when the engine was at rest, and the conditions at the weir were carefully studied to see that there were no sources of error in the measurement of the weir discharge.

At the close of the test there was a leakage through the delivery valves when the pump was at rest, amounting to about 1 per cent. of the total discharge.

The pump valves had been examined before the trial, but the water of the Ohio River carries a large amount of silt, which rapidly wears

the faces of the valves and valve seats, and after a careful consideration of the question, we concluded that the slip, as determined by the weir measurement, was probably correct within 1 or 1½ per cent.

The Boiler Performance.

The boiler performance was unsatisfactory with either kind of coal. While that with the Pittsburg coal was perhaps as good as could be expected, considering the calorific value of the coal and its capacity for covering the heating surfaces with soot, something better than the result obtained was looked for with Pocahontas coal. The latter coal was a poor sample of its kind, as shown by the analysis herewith given, and an inspection of the coal corroborated this opinion. Nevertheless, the efficiency of the boilers with Pocahontas coal was lower than should be expected with an internally fired boiler of any kind. This is explained in several ways, the principal reason being, doubtless, the fact that its use followed that of the Pittsburg coal without any cleaning of the boiler except the daily blowing out of tubes with a steam jet. Thus the heating surfaces were covered with soot from the Pittsburg coal, and the Pocahontas coal suffered in consequence. This is borne out by the higher temperature of the escaping gases. Another cause of low efficiency with Pocahontas coal is the fact of its being the first coal of that kind ever used by the firemen.

Conclusion.

The duties, steam consumption, mechanical and thermodynamic efficiencies of Pumping Engine No. 3, as given in the preceding pages, are remarkable in establishing this as the most economical compound or double expansion steam or pumping engine that has ever been tested, so far as we are aware. It is furthermore remarkable in its mechanical efficiency, as shown by the small friction of the machinery, and this contributes considerably to the duty.

The pump performance, as shown by the pump indicator diagrams, is remarkable for freedom from shocks, although there is quite an audible shock in the pumps when the discharge valves close. This is doubtless due in part to the fact that the valves and seats are wholly of metal, and in part to the sounding-board effect of the enormous mass and surface of iron of the pumps and the wooden caisson on which they rest.

The action of the steam valves and valve gear are excellent, and the same is true of all parts of the mechanism.

The results given in this report show that all requirements of the guarantee of this pumping engine are much more than fulfilled.

It is with great satisfaction that we are able to make this statement without qualification, and to further state that the whole plant is a great credit to the designers and builders.

Respectfully submitted,

DEXTER BRACKETT,
Expert for the I. P. Morris Co.

F. W. DEAN,
Expert for the Louisville Water Co.

TRIAL

OF THE

Leavitt Pumping Engine

LOUISVILLE, KY.

CAPACITY 16,000,000 GALLONS IN 24 HOURS,

BY

F. W. DEAN, BOSTON, MASS.

(Member of the Society.)

Presented at the New York Meeting (December, 1894) of the American Society of Mechanical Engineers, and forming part of Volume XVI of the Transactions.

DCXIX*

TRIAL OF THE LEAVITT PUMPING ENGINE, AT LOUISVILLE, KY., CAPACITY, 16,000,000 GALLONS IN 24 HOURS.

BY F. W. DEAN, BOSTON, MASS.

(Member of the Society.)

In April of the present year the writer, as expert for the Louisville Water Co., Louisville, Ky., and Mr. Dexter Brackett, as expert for the builders (the I. P. Morris Co., of Philadelphia) of the new pumping engine at Louisville, Ky., conducted a contract trial of six day's duration. The engine ran 144 hours and 10 minutes without a stop, which is the longest test run on record, and established itself as the most economical compound engine that has ever been tested, so far as the writer knows. The result is phenomenal and is of great interest at the present time on account of tests of some recent high expansion engines with cylinder ratios of 7 to 1, an account of one of which the writer gives in another paper. It also has great interest in showing how closely reached by this engine are the records of many triple expansion engines. The writer believes, however, that a triple expansion engine designed on the same lines will lower the steam consumption by a paying percentage.

The engine referred to is Pumping Engine No. 3, of the Louisville Water Co. It is of the well-known Leavitt type, having two vertical inverted cylinders, the piston rod of the high-pressure cylinder being connected by links to one end of a beam, and the

*Presented at the New York meeting (December, 1894) of the American Society of Mechanical Engineers, and forming part of Vol. XVI, of the *Transactions*.

low-pressure similarly to the other end of the beam. The main shaft is at one end of the engine, and the connecting rod passes from a pin in the upper part of the beam to the crank-pin. The steam pistons have opposite motions in consequence of this arrangement, and the exhausts from the ends of the high-pressure cylinder pass to the corresponding ends of the low-pressure cylinder. There are two reheating receivers between the cylinders composed of small brass tubes, inside of which is live steam of boiler pressure, the exhaust steam passing in contact with the outsides of the tubes. Both cylinders are steam-jacketed on heads and sides with steam of boiler pressure.

Each steam-cylinder is provided with four gridiron valves operated by Leavitt cams. The point of cut-off in the high-pressure cylinder is automatically determined by a ball governor, but that of the low-pressure cylinder is fixed. The engine is of the most massive character, the weight being far greater than that of any other pumping engine of the same capacity. The pumps are located directly under the engine, and the plungers are connected to the beam at such points that, while the stroke of each steam piston is 10 feet, that of each pump plunger is 7 feet. The plungers work vertically and are of the differential type, being single acting on the suction, and double acting on the discharge. The engine is provided with a surface condenser and vertical double acting air pump.

On account of the rise and fall of the Ohio River the bed plate of the engine is placed above the highest high water mark, while the bottoms of the pumps are sufficiently low to take water at the lowest stages of the river. The distance from the bottoms of the pumps to the bottom of the bed plate is 61 feet.

The trial consisted of ascertaining the duty by weir measurement at the reservoir and nearly or quite all other data of interest. That part of the trial relating to the engine only will be here described. The engine is worked by steam of 140 pounds gauge pressure at the boilers, and this is conducted through 180 feet of steam pipe, well covered, to the engine. At the engine the total per cent. of condensation in this pipe and priming of the boilers amounted to $2\frac{58}{100}$ per cent., and all of this but $\frac{55}{100}$ of 1 per cent. was thrown out by a separator. The steam pressure at the engine near the high-pressure cylinder fell to 137 pounds by gauge.

At the beginning of the trial the steam pressure in the two boilers used was at about 90 pounds, and just before starting the engine the water-level was marked in both boilers. Immediately after stopping the engine, 6 days 10 minutes later, the same pressure and water-levels existed.

From the total weight of steam entering the steam-pipe there have been deducted the steam used by the calorimeter and the water removed by the separator. In the appropriate places the moisture shown by the calorimeter was deducted, viz., wherever results are stated in terms of dry steam.

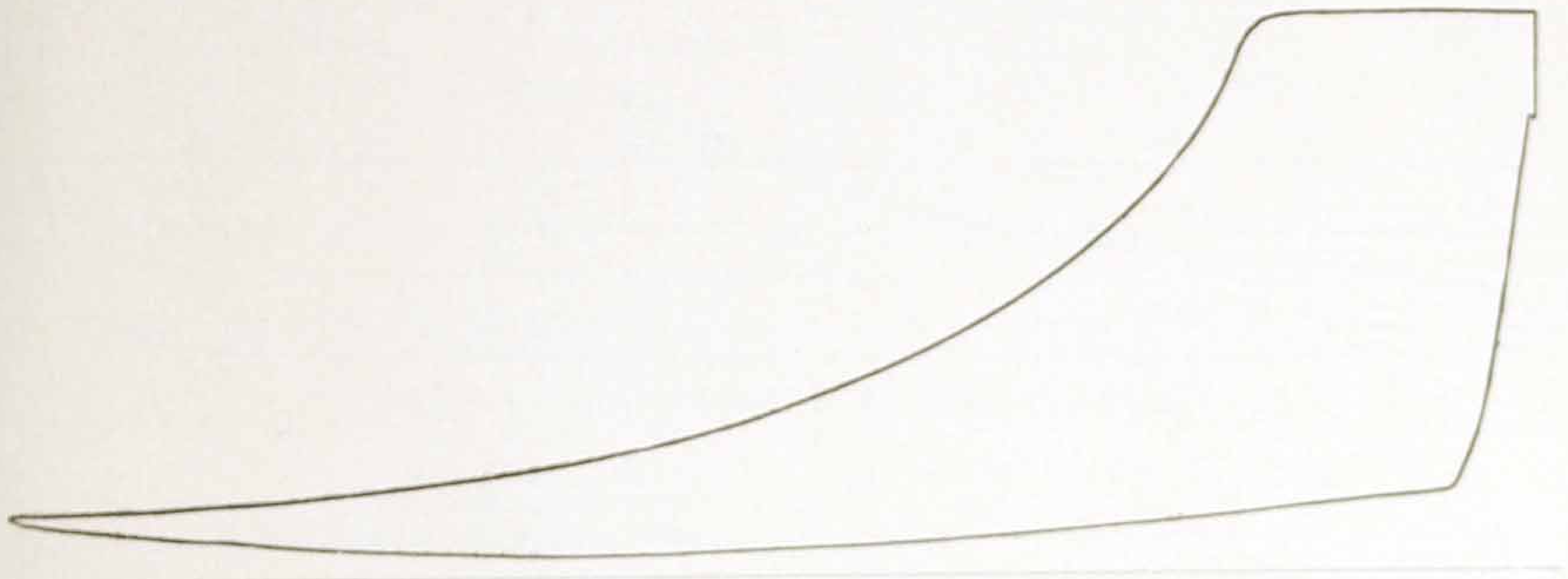


FIG. 50.—No. 123. High-Pressure, Bottom. $A = 2.78$.

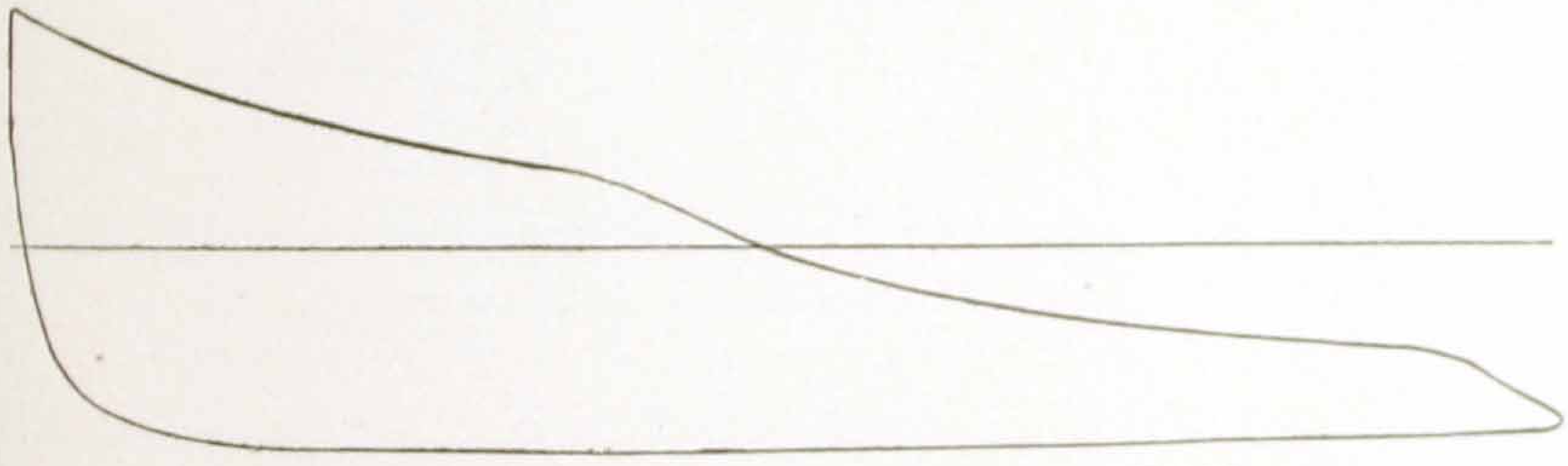


FIG. 51.—No. 123. Low-Pressure, Bottom. $A = 2.81$.

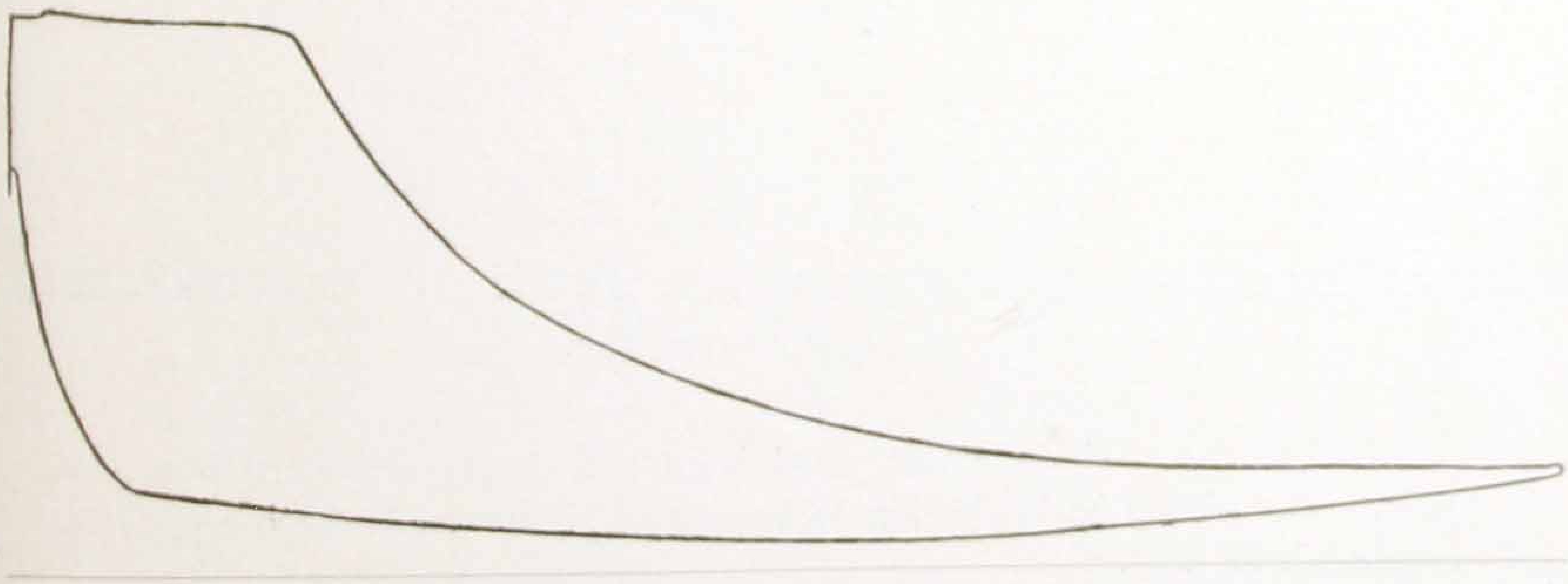


FIG. 52.—No. 123. High-Pressure, Top. $A = 2.72$.

Indicator diagrams were taken every hour throughout the 144 hours on separate indicators at each end of each cylinder. These diagrams were taken with great care, and after the trial the scales of the indicator springs were determined at the Brooklyn Navy Yard.

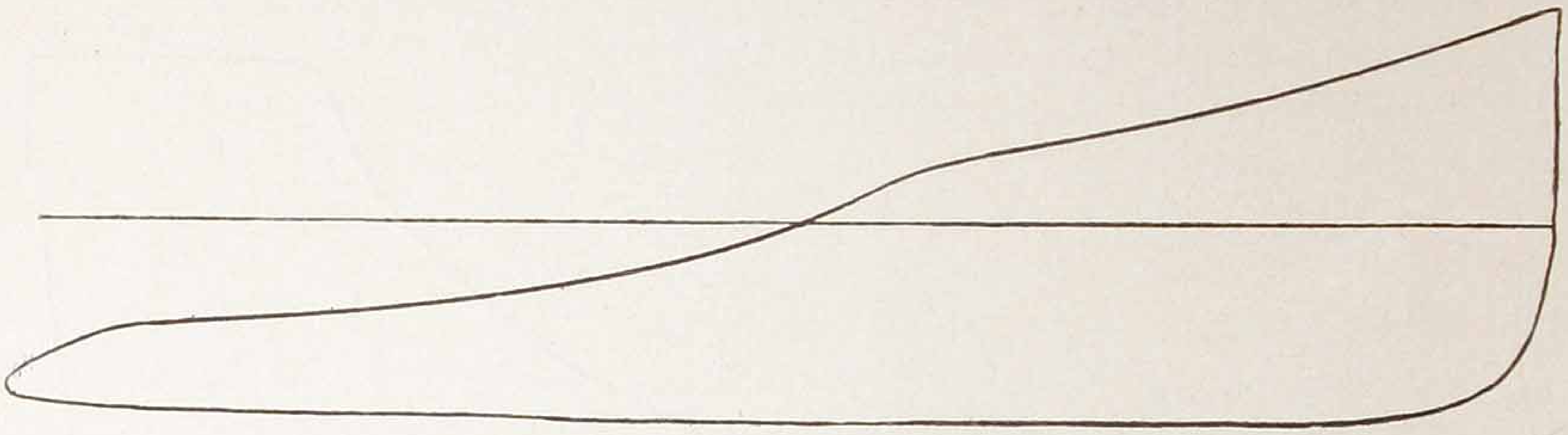


FIG. 53.—No. 123. Low-Pressure, Top. $A=2.85$.

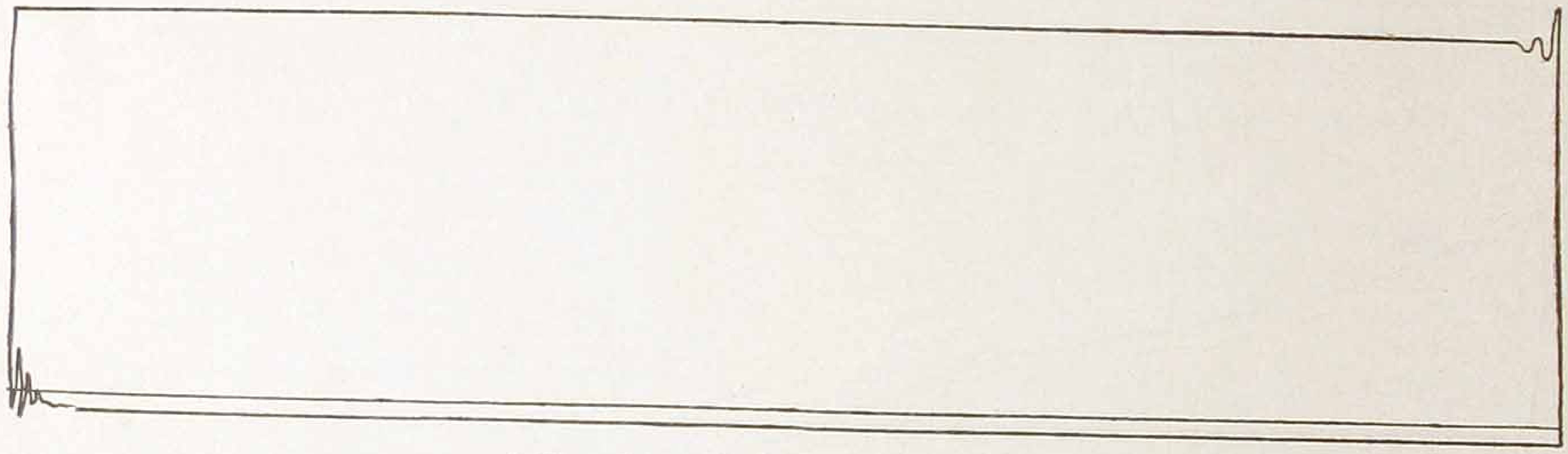


FIG. 54.—High-Pressure, End.

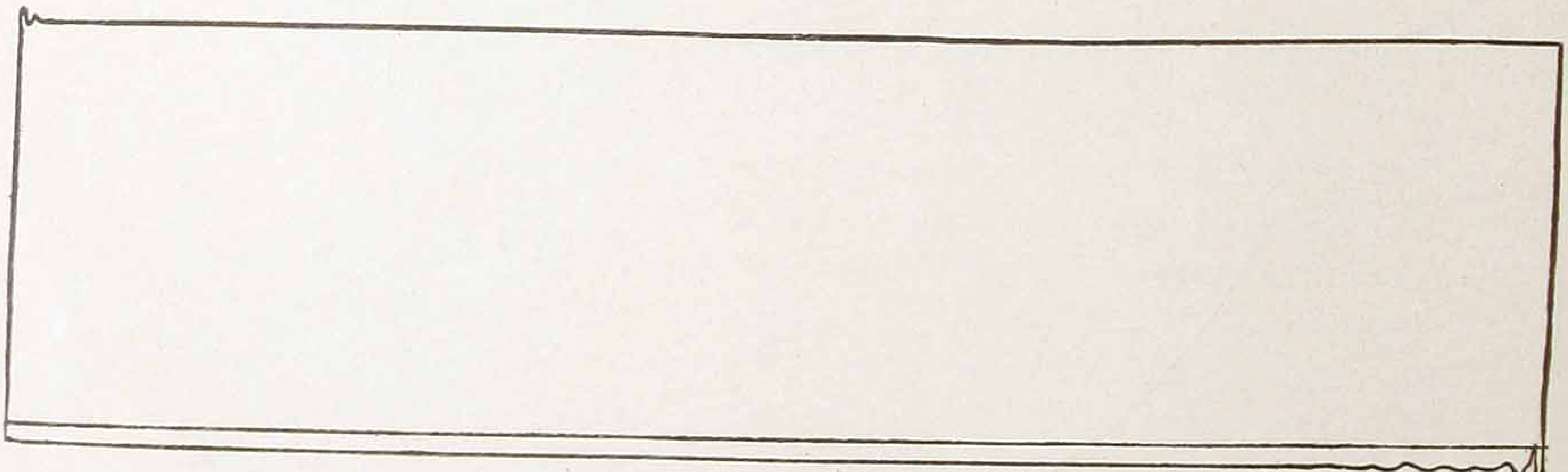


FIG. 55.—Low-Pressure, End.

Before the trial the engine had been run the greater part of a year. The piping and cylinders were thoroughly covered with a non-conductor of heat.

The following are the leading dimensions of the engine:

Type, Leavitt compound vertical inverted beam fly-wheel.

Diameter of high-pressure cylinder, hot	27.21 in.
" " low-pressure " "	54.13 in.
" " fly-wheel	36 ft.
" " high-pressure piston rod	$5\frac{1}{2}$ in.
" " low-pressure " "	6 in.
Stroke of each piston	10 ft.
Mean clearance of high-pressure cylinder	1,585 %
" " " low-pressure "	1,530 %
Diameters of each differential plunger	34 in., and $24\frac{1}{16}$ in.
Stroke " " " "	7 ft.
Mean ratio of steam piston areas	4,015 to 1.
Volume displaced by plungers during one revolution of engine	660.30 gallons.
Diameter of each discharge pipe	24 in.

Results of Engine Trial.

Duration	144 hrs. 10 min.
Total number of revolutions	160,666.5
Average number of revolutions per minute	18.574
" piston speed per minute	371.48 ft.
" plunger speed per minute	260.04 ft.

Average Temperatures.

Of Engine room	60° to 86°
Of external air	48° to 86°
Of main feed at weighing tank	81.2°
Of " " on entering boiler	108°
Of jacket and reheater drain at boiler	328.3°
Of mixture of feed-waters	143.3°
Of water in pump well	58.7°

Average Pressures.

Of atmosphere by barometer	14.60 lbs.
Of steam at boilers by gauge	140.00 lbs.
" " " " absolute	154.60 lbs.
" " " engine by gauge	137.00 lbs.
" " " " absolute	151.60 lbs.
Of initial steam, high-pressure cylinder, absolute	145.75 lbs.
Of terminal pressure, low-pressure cylinder, absolute	7.32 lbs.
Of back pressure, low-pressure cylinder, absolute	0.95 lbs.
Vacuum by gauge	27.75 in.
Water pressure by mercury column	62.50 lbs.
Height of mercury zero above water in pump well	49.04 ft.
Total water pressure	83.74 lbs.
Equivalent head	193.35 ft.

Steam used by Engine.

Moist steam entering steam-pipe	1,157,923 lbs.
Water drained from separator	23,428 lbs.

Steam used by calorimeter	727 lbs.
Total moist steam used by engine	1,133,768 lbs.
Percentage of moisture in steam after leaving separator	0.55%
Total dry steam used by engine	1,127,533 lbs.
" moist " passing through inner steam cylinders	943,973 lbs.
" " " " " steam-jackets and reheaters	189,795 lbs.
Percentage of moist steam used by jackets and reheaters	16.74%
Moist steam used per hour, per I. H. P.	12.223 lbs.
Dry " " " " " "	12.156 lbs.
" " " " " " " by inner clyinders	10.120 lbs.
Moist " " " " " pump, horse-power	13.125 lbs.
Dry " " " " " " " "	13.050 lbs.
Prevailing point of cut-off high-pressure cylinder	20.20%
" " " " low-pressure "	42.10%
Drop between cylinders	0.00 lbs.
Compression in high-pressure cylinder	full.
" " low-pressure "	5/8 full.
Ratio of expansion by volume	20.40
Steam accounted for by indicator at high-pressure cut-off in per cent. of 10.120 pounds	7.75 lbs. = 76.58%
Steam accounted for by indicator at high-pressure release	9.166 lbs. = 90.57%
" " " " " low-pressure cut off	10.008 lbs. = 99.60%
" " " " " " " release	9.725 lbs. = 96.09%

NOTE.—The last four items are to be regarded as closely approximate only.

Average Powers, Etc.

Average mean effective pressure in high-pressure cylinder	43.53 lbs.
" " " " " low-pressure "	14.155 lbs.
Horse-power developed by high-pressure cylinder	279.00 H. P.
" " " " low-pressure "	364.40 "
" " " " both cylinders	643.40 "
Percentage of power in high-pressure cylinder	43.36%
" " " " low-pressure "	56.64%
Horse-power of plungers	599.10 H. P.
Friction horse-power	643.40–599.10 = 44.30 "
Efficiency of mechanism	93.12%
Friction of mechanism	6.88%

British Thermal Units, Etc.

Mean absolute steam pressure at engine	151.60 lbs.
" temperature of rejection of engine (air pump discharge)	84.2°
Mean temperature of rejection of jackets and reheaters at engine	335.3°
Heat of liquid of air-pump discharge	52.27 B. T. U.
" " " " jacket and reheater drain	306.00 "
" " vaporization of steam supply	860.60 "
" " liquid of steam supply	330.90 "

Dry steam in mixture used by engine	0.9945
B. T. U. per lb. of moist steam passing through inner cylinders, $0.9945 \times 860.6 + 330.90 - 52.27 =$	1134.50 B. T. U.
B. T. U. per lb. of moist steam passing through jackets and reheaters, $0.9945 \times 860.6 + 330.9 - 306.0 =$	880.8 "
B. T. U. passing through cylinders in 144 hrs. 10 min.	1,070,937,531 "
B. T. U. passing through jackets and reheaters in 144 hrs., 10 min.	167,171,428 "
B. T. U. passing through engine in 144 hrs., 10 min.	1,238,108,959 "
B. T. U. used per I. H. P. per minute (moist steam)	222.46 "
Mechanical equivalent of heat (Rowland)	778 ft. lbs.
Thermodynamic efficiency of engine $\frac{33000}{222.46 \times 778} =$	19.07%

Duties based upon Plunger Work.

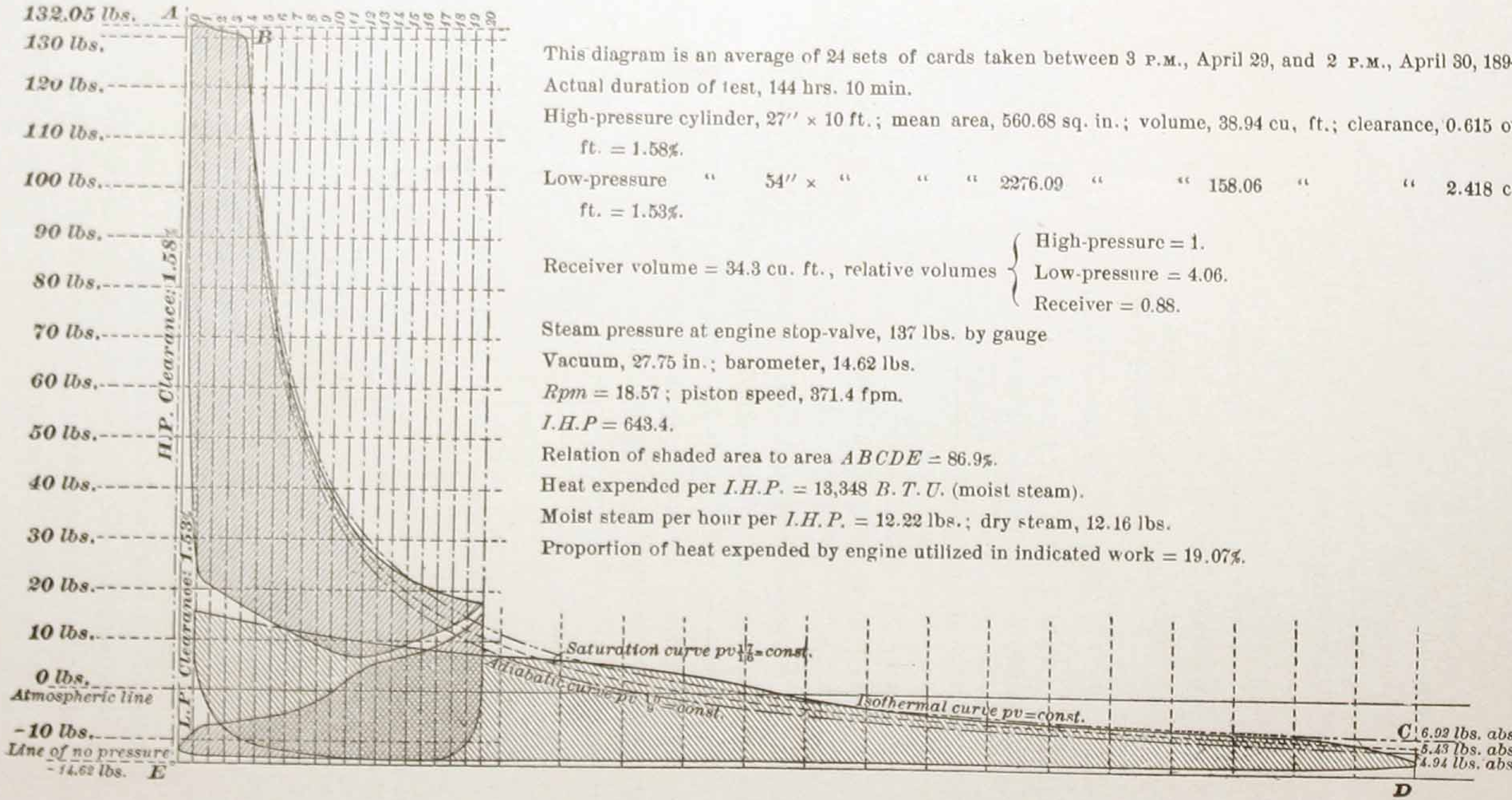
Plunger work performed in 144 hrs., 10 min.	171,015,314,960 ft. lbs.
Duty per 1,000,000 B. T. U. used by engine alone	138,126,000 "
" " 1,000 lbs. moist steam used by engine alone	150,838,000 "
" " 1,000 lbs. dry steam used by engine alone	151,672,000 "
" " 100 " " Pittsburg coal	125,444.000 "
" " 100 " " Pocahontas "	139,031,000 "
" " 100 " " Pittsburg combustible	129,295,000 "
" " 100 " " Pocahontas "	145,762,000 "

Sample indicator diagrams from steam and water cylinders are given (Figs. 50–55), and also a combined diagram (Fig. 56).

This engine is, both in design and results, in striking contrast with the Rockwood System engine described in the writer's other paper, as shown in the following table:

ENGINE.	LEAVITT.	ROCKWOOD.
Steam pressure absolute	151.60 lbs.	175.50 lbs.
Vacuum	27.75 in.	25.3 in.
Ratio of expansion	20.40	33.00
Number of revolutions per minute	18.57	76.4
Length of stroke	10 ft.	4 ft.
Piston speed per minute	371.5 ft.	611.2 ft.
Cylinder ratio	4 to 1.	7 to 1.
Drop between cylinders	None.	About 14 lbs.
Dry steam per I. H. P. per hour	12.156 lbs.	12.84 lbs.
Difference in favor of Leavitt	0.684 lbs. = 5.3%	

This comparison shows very clearly that the ratio of 7 to 1 does not necessarily produce as economical results as a ratio far removed from it, even with the additional advantages of 24 pounds more steam pressure, 1.6 times as many expansions, four times as many reciprocations per minute, and twice as great piston speed. It tends to show



This diagram is an average of 24 sets of cards taken between 3 P.M., April 29, and 2 P.M., April 30, 1894:

Actual duration of test, 144 hrs. 10 min.

High-pressure cylinder, 27″ × 10 ft.; mean area, 560.68 sq. in.; volume, 38.94 cu. ft.; clearance, 0.615 cu. ft. = 1.58%.

Low-pressure " 54″ × " " " 2276.09 " " 158.06 " " 2.418 cu. ft. = 1.53%.

Receiver volume = 34.3 cu. ft., relative volumes { High-pressure = 1. Low-pressure = 4.06. Receiver = 0.88.

Steam pressure at engine stop-valve, 137 lbs. by gauge

Vacuum, 27.75 in.; barometer, 14.62 lbs.

Rpm = 18.57; piston speed, 371.4 fpm.

I.H.P = 643.4.

Relation of shaded area to area *ABCDE* = 86.9%.

Heat expended per *I.H.P.* = 13,348 *B.T.U.* (moist steam).

Moist steam per hour per *I.H.P.* = 12.22 lbs.; dry steam, 12.16 lbs.

Proportion of heat expended by engine utilized in indicated work = 19.07%.

FIG. 56.

that no advantage arises from a drop in pressure between the cylinders, if evidence were needed of this.

It is the writer's opinion that in order to use steam in the most economical manner in a multiple expansion engine, the expansion must be continuous throughout the series of cylinders (that is to say, there should be no drop between the cylinders), and that compression should be carried up to the initial pressure in each cylinder. These features have been employed to the fullest extent in the Leavitt engine which forms the subject of this paper, and the result has surpassed all the records for economy of engines of its class

ADDED TO THE PAPER AFTER THE MEETING.

A test of so much importance as that of the Louisville engine, wherein a new record has been established for steam consumption, may, with great propriety, be accompanied by some data of the log of the trial.

The following are the amounts of feed-water weighed each day of twenty-four hours :

1st day, 24 hours	159,752 pounds of water.
2d " " "	161,799 " " "
3d " " "	159,972 " " "
4th " " "	161,848 " " "
5th " " "	160,438 " " "
6th " 24 hours 10 minutes	164,439 " " "

JACKET AND RE-HEATER RETURN METER READINGS.

TIME.	Reading.	Difference.	Weight each cubic unit.	Total weight.
2.55 P. M., April 25, 1894	9684.5	......	....	....
3 " " " "	9690.8	6.3	52.17 lbs.	329 lbs.
" " " 26, "	10290.0	599.2	" "	31,260 "
" " " 27, "	10884.0	594.0	" "	30,989 "
" " " 28, "	11493.0	609.0	" "	31,771 "
" " " 29, "	12097.0	604.0	" "	31,510 "
" " " 30, "	12708.0	611.0	" "	31,876 "
3–1 " May 1, "	13319.3	611.3	" "	31,891 "
3–10 " " " "	13322.5	3.2	" "	167 "

The above unit weight of 52.17 pounds was determined by weighing the condensation in a cask of cold water during two hours. Thinking that this calibration might be too short to use for a test of

six days' duration, a calibration of twenty-four hours' duration was made by Mr. Hermany, chief engineer and superintendent of the Louisville Water Works, at my request. The engine was run at precisely the same speed and against the same head as existed during the official trial. Indicator diagrams were taken every hour, worked up by me, and gave the same horse-power as on the official trial. During these twenty-four hours the condensation was continuously weighed, and found to be 31,732½ pounds, which is almost identical with the meter results. During this same trial the feed-pump was run by a donkey boiler, and its exhaust was not allowed to enter the boiler. The weighed feed amounted to 193,133 pounds, which is almost identical with the sum of the weighed feed and jacket and re-heater returns, as given above, for any single day.

On October 10 the condensations in the jackets and re-heaters were determined by weighing separately, but simultaneously, during eight hours, with the following results:

Average Condensations per Minute, October 10.

High-pressure Jacket.	Re-heaters	Low-pressure Jacket.
7.4938 pounds.	10.0417 pounds.	5.2333 pounds.

On October 20 the same determinations were repeated for eight hours, with the following results:

Average Condensations per Minute, October 20.

High-pressure Jacket.	Re-heaters.	Low-pressure Jacket
7.5083 pounds.	9.9437 pounds.	5.1417 pounds.

The engine ran on each of these trials at the following speeds:

Average number of revolutions per minute, October 10...............18.6083,
" " " " 20...............18.5979,

while on the official trial the average number of revolutions per minute was 18.574. It is not known what head existed on either October 10 or October 20, and, therefore, what power was being generated.

[NOTE.—This paper received discussion jointly with that by the same author on "Trials of a Recent Compound Engine with a Cylinder Ratio of 7 to 1," and the remarks made in debate are published in connection with that paper.]

DCXX.*

TRIALS OF A RECENT COMPOUND ENGINE WITH A CYLINDER RATIO OF 7 TO 1.

BY F. W. DEAN, BOSTON, MASS.

(Member of the Society.)

CONSIDERABLE interest has been recently shown in the performances of some compound engines working with high pressure steam; and members will recall a paper presented at the San Francisco meeting by Messrs. Green and Rockwood, giving an account of trials of an engine as a triple expansion engine and, by throwing the intermediate cylinder out of use, as a compound.† The results of the trials, which were evidently made with due care, tended to establish equal economy of the two types.

Laying aside for the present consideration of the possibility of such results being obtained from well-designed and properly worked engines of the two types, the writer desires to give an account of a test which he conducted of an engine founded, in its design, upon the engine referred to, and embodying what is known as the Rockwood system.

This engine was built by the Wheelock Engine Company, of Worcester, Mass., for B. B. & R. Knight, of Providence, R. I. and located at their mill in Natick, R. I. The engine possessed the cylinder ratio of 7 to 1, which, under the system referred to, is held to possess special virtue.

* Presented at the New York meeting (December, 1894) of the American Society of Mechanical Engineers, and forming part of Volume XVI of the *Transactions*.

† *Transactions American Society of Mechanical Engineers*, Vol. XIII, p. 647, No. 499.

The following are the leading dimensions:

Diameter high-pressure cylinder, hot	18.44 in.
" low-pressure " "	48.50 in.
" high-pressure piston-rod	3.25 in.
" low-pressure "	4.25 in.
Stroke of both pistons	48.00 in.
Mean ratio of piston areas	7 to 1
" high pressure clearance	$2\frac{3}{4}$%
" low pressure "	$2\frac{1}{2}$%

The engine is a horizontal cross compound, with the high-pressure cylinder jacketed all over, and the low-pressure cylinder on the heads only. There was a re-heater between the cylinders. In the writer's judgment the jackets were badly piped, and it is doubtful if the jacket circulation was good. The re-heater was quite deficient in heating surface. The condenser was of the injector type, made by the builder of the engine. The vacuum was defective, although very cold water was used.

The engine was four hundred feet from the boiler, which was of the Babcock & Wilcox make, but as the pipe and flanges were well covered the condensation was not excessive.

Examination showed the pistons and valves to be tight.

The feed water was weighed upon correct scales, and was pumped by a geared pump. The boiler was entirely separate from others in the same plant, and all connected pipes which could carry unaccounted for water or steam to or from the plant were disconnected or blanked. There were no leaks either in the economizer or boiler, and in the second test here described the economizer was not in use.

In the engine room, indicator diagrams were taken by two indicators on each cylinder every twenty minutes, the power being very uniform. A calorimeter was attached to the main steam-pipe near the high-pressure cylinder, and just before it there was located a steam separator. The condensation from this separator was kept at a constant height in a water glass, and the water drawn off was weighed by running it into a tank of cold water. The re-heater and jacket condensations were under control, and were kept at a visible and constant height in a glass tube, thus insuring no waste of steam.

Five different tests were made, but on account of accidental and unavoidable wastes of steam in three of them, only two will be quoted here. During the two referred to there was a slight leak of steam from an expansion joint, and on the last test one safety valve was open three-quarters of a minute. These errors are so slight that they can be ignored.

The indicator springs were carefully tested by the writer under steam, and afterward taken to the Navy Yard at Brooklyn and tested, the two results being substantially alike.

The durations of the tests were shorter than is desirable, but the mill hours determined this.

The following is a brief tabulation of the results:

Date, 1894.	JAN. 26, P. M.	JAN. 27, A. M.
Duration of trials	$4\frac{1}{2}$ h.	5 h.
Average steam pressure near engine	159 lbs.	158 lbs.
" vacuum	25.4 in.	25.2 in.
" ratio of expansion by volumes	33.0	33.4
" number of revolutions per minute	76.357	76.603
" piston-speed, feet per minute	610.86	612.82
Per cent. of moisture in steam near cylinder	1.90%	1.75%
Total dry steam used	34.089 lbs.	37.677 lbs.
Average I. H. P.	594.79	582.21
Dry steam used per I. H. P. per hour	12.74 lbs.	12.94 lbs.
Average dry steam used per I. H. P. per hour	12.84 lbs.	

It will be seen that these results show a very economical use of steam, and far less than has heretofore been thought possible with compound engines. If the vacuum had been 28 inches, the steam consumption might have been as low as 12.36 pounds on January 26, P. M.,

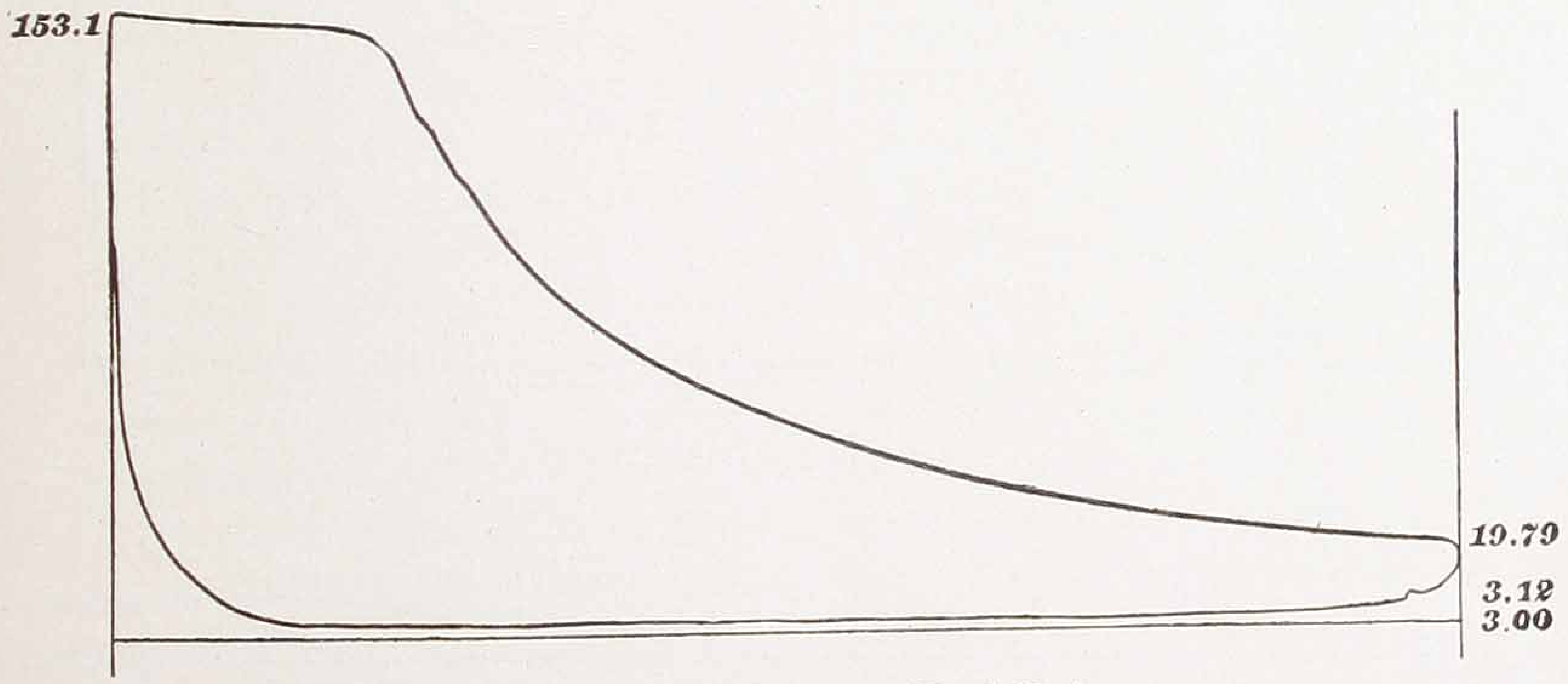


FIG. 57.—High-Pressure, Head End.

and 12.60 pounds on January 27, A. M., if this had not given rise to any unfavorable set of thermodynamic conditions. The average of these two is 12.48 pounds.

Sample indicator diagrams are given (Figs. 57–60,) and in the

writer's opinion they have a grave defect in showing a considerable drop in pressure between the cylinders. The writer is aware that this is desired by the designers, but the loss in effect of the steam to which

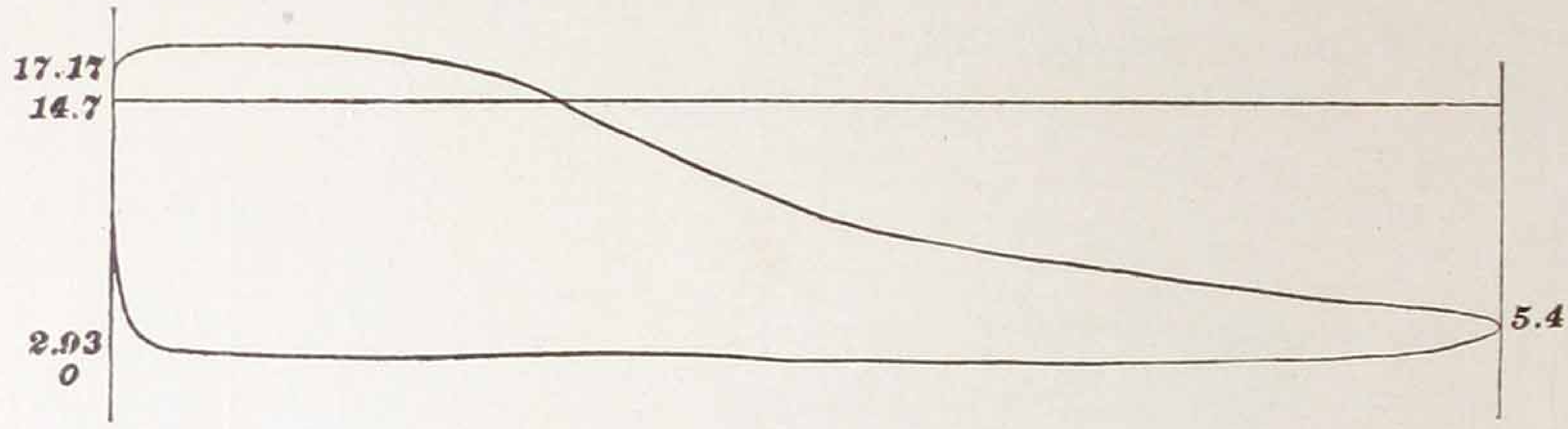


FIG. 58.—Low-Pressure, Head End.

this gives rise cannot be recovered by any subsequent event. Moreover, this drop exaggerates the difference in the ranges of temperatures of the two cylinders, and increases the loss still further by increasing the cylinder condensation, according to the well-known and funda-

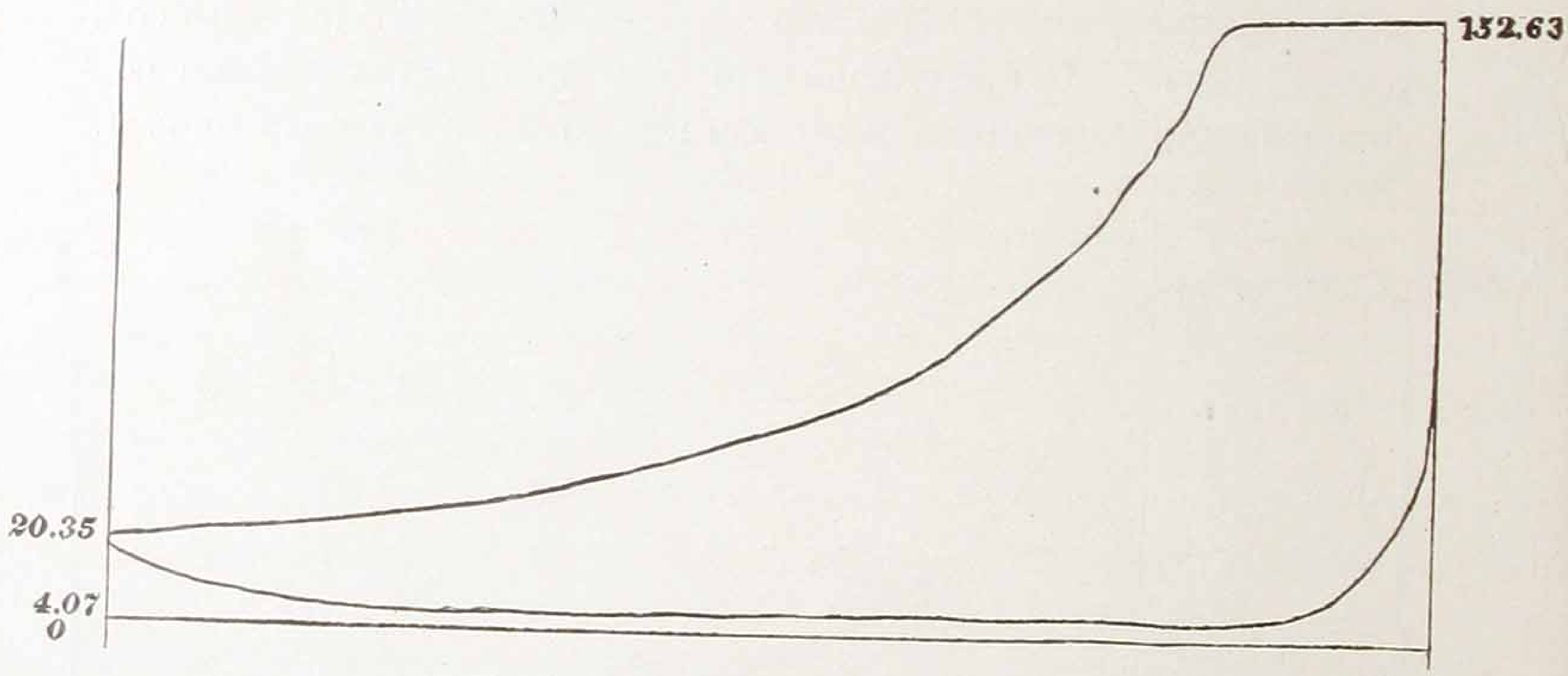


FIG. 59.—High-Pressure, Crank End.

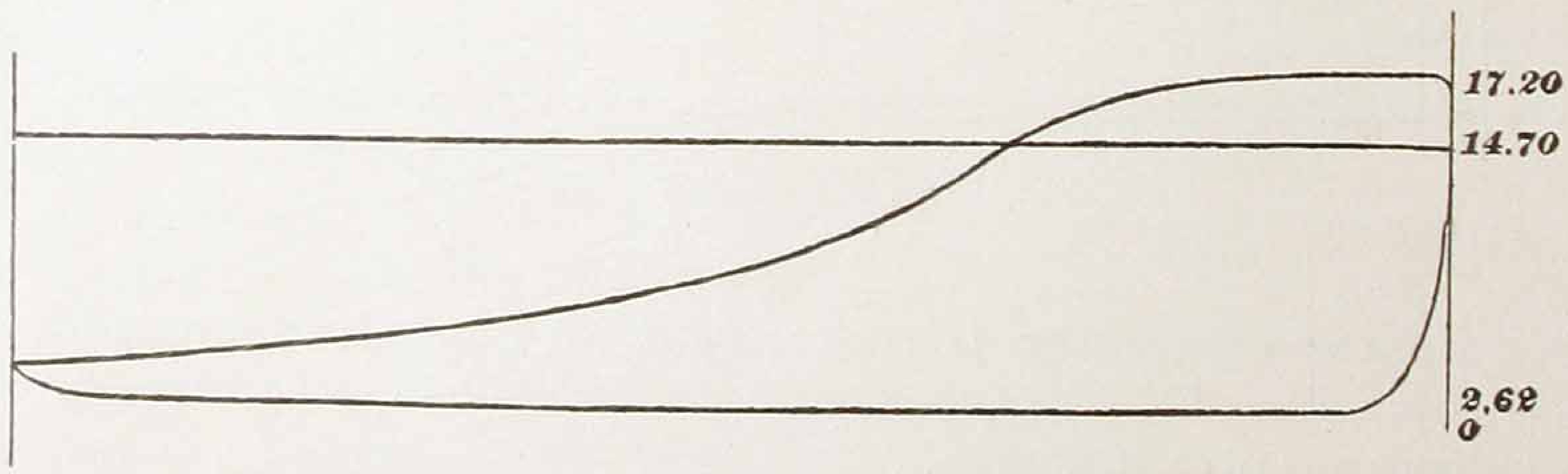


FIG. 60.—Low-Pressure, Crank End.

mental theory of the desirability of equal ranges of temperatures. The ranges on January 27 were about 144 degrees in the high-pressure and 82 degrees in the low-pressure cylinders.

Although the performance of the engine is remarkably good the writer believes that it was realized in spite of great defects, and that it would have been much better if these alleged defects had not existed. The economy, in the writer's judgment, is due to high steam pressure with the resultant high degree of expansion, small clearances, and tight pistons and valves.

DISCUSSION.

Prof. R H. Thurston.—The results of short engine trials have always been looked upon with much distrust by engineers, when apparently exhibiting exceptional economy; and the traditional myth of the performance of the "record-breaking" Cornish engine of the last generation, and of that of the S. S. *Thetis*, in which, for the time, fabulous duties are stated as the results of short duty-trials by famous engineers, are a standing admonition to all later experimenters. This reproach certainly cannot be urged against this trial, and the profession is placed under a real obligation to Mr. Leavitt, Mr. Dean, and Mr. Hermany for the admirable example which they have here given of deductions based upon unquestionably representative and usual extended periods of operation under regular working conditions. The machine should, it is fair to assume, be capable of sustaining this duty indefinitely. A week's work should be as satisfactorily representative of the capabilities of the engine as the work of a year. In this case, the result is a magnificent one, and designer, builders, and officers in charge of the machine have reason for congratulation. I think this "breaks the record" for the compound engine to date. A duty of 140,000,000 for 100 pounds of coal, and of above 150,000,000 for 1,000 pounds of steam, represents, probably, not only the best to date for this class of engine, but, very closely, the practical limit with saturated steam; and 12 pounds of steam per I. H. P. per hour seems the limit for pressures of 125 to 150 pounds.

The usual conditions of economy are here illustrated fully: dry steam, sharp cut-off. full expansion to six or eight pounds absolute pressure, free transfer of heat in jackets, with small cylinder-condensation, no drop between cylinders, and high efficiency of mechanism. The jacket-condensation is high, the friction of mechanism extraordinarily low—for such a heavy engine very remarkably so. I doubt if it

has ever been equalled, except by the Worthington class of direct-acting machines.

COMPOUND *vs.* TRIPLE.

ENGINE.	LEAVITT.	ROCKWOOD.	TRIPLE.
Steam pressure absolute...............	151.60 lbs.	175.50 lbs.	135.45 lbs.
Vacuum....................................	27.75 ins.	25.3 ins.	27.6 ins.
Ratio of expansion........................	20.40	33.00	19.55
Number of revolutions per minute...	18.57	76.4	20.31
Length of stroke.........................	10 ft.	4 ft.	5 ft.
Piston speed per minute...............	371.5 ft.	611.2 ft.	203 ft.
Cylinder ratio..............................	4 to 1.	7 to 1.	1, 3, 7
Drop between cylinders.................	None.	About 14 lbs.	....
Dry steam per I. H. P. per hour.....	12.156 lbs.	12.84 lbs.	11.678 lbs.
Difference in favor of Leavitt.........	0.684 lbs. = 5.3%	....	....
" " " " Triple..........	0.478 lbs. = 4%	1.16 = 9%	....

As presenting an interesting comparison, I have taken the liberty of adding to Mr. Dean's table the figures for the Milwaukee engine, in order to bring especially the relation of compound to triple, as a comparison of the best work in each class permits now, with a conclusiveness never before allowed. In the collection of data thus assembled, we find the triple expansion machine with lowest steam pressure, lowest piston speed, and lowest ratio of total expansion, gives four per cent. higher economy than the compound, nine per cent. better than the hermaphrodite machine, and this means, no doubt, that Mr. Dean's statement is perfectly correct; that the triple engine would have proved in the hands of Mr. Leavitt as remarkable in its class as is this compound in its field. The observed difference would be exaggerated, were the triple given the advantage of equally high steam and high piston speed, and it would seem probable that, under the conditions here indicated, the gain by the addition of the third cylinder would be something over five per cent. The loss by leaving it out, and substituting a receiver with free expansion, would seem, under similar conditions, to be likely to prove in excess of ten per cent.; a high price to pay for the saving of even a steam cylinder with its valve-gearing. Mr. Leavitt's success is one in which the whole profession may find cause for pride.

Mr. F. H. Ball.—This paper institutes certain comparisons between the Leavitt Pumping Engine at the Louisville Water Works, and another engine which is described as the "Rockwood System," and

certain deductions are made by the author as a result of these comparisons. Unfortunately for some of us, at least, who are interested in this subject, we have not been informed as to exactly what the "Rockwood System" is. We have had several very interesting reports from Mr. Rockwood, of trials of compound engines, where cylinder ratios larger than usual were used, and many of us, who believe he is on the right track have hoped that he would elaborate his theory in a paper for presentation to this Society. If the ratios commonly used are wrong, there must be some theory to demonstrate this fact, and to point to some other ratio as being better. Mr. Rockwood has told us, on different occasions, of his engines, with various cylinder ratios, ranging as high in one case, I believe, as 9 to 1. Does his system then consist of simply making cylinder ratios greater than heretofore, and does it cover all cases from the conventional ratio to infinity, or is there a choice in this matter? Mr. Dean seems to think that he has located Mr. Rockwood at 7 to 1. Let us proceed on this assumption.

Referring to the performance of the two engines under consideration, it must be admitted that the results obtained in both cases are phenomenal. Here are two compound engines showing an economy that has seldom been equalled by the best triple-expansion engines, and never exceeded by them but by a very small amount. The Leavitt engine stands at the head, with its 12.15 pounds of water per horse-power per hour, and the Rockwood engine a good second with 12.84 pounds. In comparing these remarkable engines, Mr. Dean has made some sweeping conclusions, that perhaps may be fairly questioned.

On the last page of his paper he uses the following language:

> "It tends to show that no advantage arises from a drop in pressure between the cylinders, if evidence were needed of this."

Also, in the closing paragraph, Mr. Dean says:

> "It is the writer's opinion that in order to use steam in the most economical manner in a multiple expansion engine, the expansion must be continuous throughout the series of cylinders (that is to say, there should be no drop between the cylinders,) and that compression should be carried up to initial pressure in each cylinder."

I must take issue squarely with Mr. Dean, both in regard to this being a reasonable conclusion from the figures of his test, and also in regard to its being true.

Taking the question of compression first, where is there, in the reported data of these engine trials, one iota of evidence on the subject of compression? Here we have two engines, with widely differing

cylinder ratios, tested under conditions that are dissimilar in almost every respect. In comparing the two engines, the least conspicuous difference is in regard to their relative rates of compression. Therefore I don't think Mr. Dean is warranted in arriving at any conclusions whatever regarding compression, from the figures of these trials. If his compression theory rests on any other evidence, I hope he will give it to this society in connection with this paper. As against his theory we have the engine trials conducted by Professor Jacobus, reported at the Montreal meeting of our Society, in which trials all the conditions remained practically constant except compression, and the evidence obtained is conclusive that full compression did not in this case give the best economy. Does Mr. Dean question the accuracy of the data reported by Professor Jacobus, or if not, how does he make his theory fit these facts?

Coming back to the other part of his opinion, he tells us that "There should be no drop between the cylinders." Persumably this opinion is confirmed in his mind by a study of the data obtained in his trial of the two engines under consideration. Let us see how logical this looks.

First, the great dissimilarity of conditions governing these tests would seem to make it very difficult to estimate the effect of any one of the features of difference, because all of these differences were present continuously during the tests, and each producing its own modification of the result.

Second, let us suppose, however, that the case was different, and that the two engines were exactly alike in every respect except as to the cylinder ratios, and the consequent terminal drop. Let it be assumed also that the conditions of the test were identically the same with both engines. The Leavitt engine, Mr. Dean tells us, represents his theory to the fullest extent. This engine has a cylinder ratio of 4, and, without appreciable drop between the cylinders, maintains a practically continuous expansion to about 20 volumes.

The Rockwood engine has a cylinder ratio of 7 and a considerable terminal drop between the cylinders, and expansion is carried to about 33 volumes. Between these wide extremes there is a vast unexplored wilderness, so far as any information from these tests is concerned. If the economy of these engines was represented graphically with relation to the economy of similar engines with greater cylinder ratios than Rockwood, and less than Leavitt, the result would be a curve on which Rockwood and Leavitt would appear near that part of the curve representing the best economy, and beyond Rockwood at one end, and Leavitt at the other, the curve would turn toward a

greater consumption of steam. Suppose Mr. Dean has established two points on this curve with the data from these engines. How can he, without a third point, locate the curve, and say that Leavitt is at the lowest point? He may with propriety say that this engine shows the best recorded performance, and that it is better than the performance of the Rockwood engine which he tested, but it seems to me that he has no warrant from these figures for saying that "There should be no drop between the cylinders," nor that "compression should be carried up to initial pressure in both cylinders," because it is only a surmise on his part that a still better result than either would not have been obtained with some compromise between the two.

The net result of any engine trial is simply the combined result of a great variety of conditions, and hence the uncertainty of attributing a good result or a bad one to any one of these numerous conditions, without having carefully tested for that condition. Anything short of this is mere guess work, which we are all privileged to engage in as a diversion, but which has little value from a scientific standpoint. Mr. Dean finds a slightly better result with the Leavitt engine than the Rockwood, and guesses that it is due to full compression and no drop between the cylinders. From the standpoint of his theory he finds an unexpectedly good result from the Rockwood engine, and guesses again, "That it was realized in spite of great defects." Following Mr. Dean's example I am inclined to guess that the economy of the Leavitt engine is realized "in spite of great defects," and these defects I should call the full compression and lack of drop between the cylinders, which are the very features he commends as being the full realization of his theory. In this matter of guessing we are both now on record, and can await the verdict of future experiments. The Jacobus tests, already referred to, seem to be good evidence on the subject of compression, and if Mr. Dean has anything else in this line, he will no doubt offer it in closing the discussion of his paper.

On the question of terminal drop, my reasons for differing with Mr. Dean will be found on the following pages, which I shall be glad to have criticised and discussed by Mr. Dean, or any member of the Society.

First, assuming that, in a given compound engine, the most economical range of temperature and pressure for each cylinder is known; then is it not the function of each cylinder to effect the most economical use of steam between the extremes of pressure through which it works?

Second, considering the low-pressure cylinder alone, and assuming that a fixed receiver pressure is practically maintained, may not the

economy of the low-pressure cylinder be studied apart from the high-pressure cylinder, and is it not true that the economy or wastefulness of the low-pressure cylinder cannot affect in any manner the economy of the high-pressure cylinder under the assumed conditions as to constant receiver pressure?

Third, referring to the high-pressure cylinder, and still assuming a practically constant receiver pressure as before, is it not true that the economy or wastefulness of this cylinder produces no effect on the economy of the low-pressure cylinder, provided the low-pressure cylinder is made to account only for the steam delivered to it from the receiver?

Fourth, assuming that the foregoing questions have been answered in the affimative, let it further be assumed, for reasons that will appear later, that the boiler pressure is such that a receiver pressure equal to the atmospheric pressure has been found the most economical. Under the foregoing conditions, then, the high-pressure cylinder would perform exactly the functions of the single cylinder of a simple engine without the condenser, because it would receive steam at boiler pressure and reject it at atmospheric pressure.

This brings the subject down to a point where the writer is glad to agree heartily with Mr. Dean in his statement regarding the high-pressure cylinder, when he says that any loss in effect of the steam in this cylinder "cannot be recovered by any subsequent event." If this is true, then, for the best results from this engine, it is necessary that the high-pressure cylinder should develop the highest possible economy when receiving steam at boiler pressure and discharging it at atmospheric pressure, and, as has already been stated, this is exactly the function of the simple non-condensing cylinder; therefore the data obtained in trials of simple engines may be safely applied to the high-pressure cylinder of a compound engine such as we have under consideration. This opens for us a vast field of research among reliable records of simple engine trials, and if Mr. Dean will point to a single case where the best economy from a simple engine was obtained by expanding to atmospheric pressure, and thus eliminating terminal drop, it will greatly fortify his theory. Is it not true, that in every instance where simple engines have been tested at various points of cut-off, that the highest economy has always been found when the expansion curve terminated at a point higher than the atmospheric pressure? This terminal drop results in a loss of work, it is true, and this loss increases rapidly with increase of drop, as was illustrated in a paper which the writer presented to this Society at the Montreal meeting; but, up to a certain point, this loss is more than overcome by the

resulting increase of mean effective pressure relatively to the cylinder condensation. Terminal drop or free expansion develops heat by internal work in the steam, and thus produces a superheating effect in the steam discharged under these conditions. In the case of a simple engine, this superheating is lost by being discharged into the atmosphere, while, with the compound engine which we are considering, the low-pressure cylinder utilizes this superheat, and therefore a greater terminal drop is permissible than when the cylinder discharges into the atmosphere. For the purpose of utilizing the data obtained from trials of simple engines in this investigation, a receiver pressure equal to the atmosphere was chosen. Whatever can be shown to be true with the boiler pressure and receiver pressure, we have assumed will also be true with regard to other pressures, to some degree, at least. The foregoing course of reasoning is conclusive to my mind that Mr. Dean's theory is wrong, and it is to be hoped that this question may be definitely settled soon, by carefully conducted experiments, having that object solely in view.

Mr. George I. Rockwood.—The two papers presented by Mr. Dean naturally interest me very much, and I trust I may be pardoned if I discuss them at some length; as, though terse (and, I may add, refreshingly so,) yet they bear with force upon not only the relative thermodynamic merits of the two engines whose economic performance they describe, but also upon the general theory of the high-duty steam-engine.

Let us refer to the contrast said to exist between these two engines. Take, as the first consideration, the steam end of the Louisville engine. This may be reasonably regarded as embodying the best design, and, perhaps, the best mechanical execution that we can hope to secure in an engine having two cylinders of a volume ratio of 1 to 4, working under a steam pressure of 140 pounds, and under pumping-engine (that is, the best) conditions. These conclusions are confirmed by the news in Mr. Dean's paper of its actual performance; an inspection of the indicator diagrams shows that the thermodynamic conditions of its operation can hardly be improved.

Consider, second, the Natick compound engine, which embodies in its design the extreme cylinder volume ratio of 1 to 7; it has small clearances and large ports in the cylinders, its pistons and valves are reasonably tight, though manifestly not perfectly so, as I will presently show. It has a relatively large intermediate receiver (a very important adjunct to the engine,) which, as Mr. Dean says, contains rather too few brass tubes to produce the best steam-jacket effect, although baffle plates are used to get the utmost possible contact of steam with tubes.

In one important point the design of the engine is not on "all fours" with that of the Louisville engine; namely, it has no barrel jacket on the low-pressure cylinder.

Now I do not agree with Mr. Dean that the conditions of operation of each engine are such as to make the comparison of duties actually attained a perfectly fair one from which to judge between the relative economic advantages of the two different systems of designing, which, as machines, no doubt these engines illustrate very well. However, a pretty fair estimate can be formed if only correct inferences are drawn from the data Mr. Dean gives us. Allow me to say here that although the different parts of the Natick engine, such as the details of the cylinders, the details of the valves and valve-gears, and the running parts, and the volume of the receiver of this engine, were decided upon by myself, yet I never saw the engine but twice in my life; once, after it was erected and had been running some months, and once after it was tested. The details of its application to the place where it now is I had nothing to do with.

I think, with the author, that the jacket circulation of this engine is perhaps poor; that the re-heater does its work under adverse conditions; that the vacuum was not so good, by an amount which I estimate from the papers at 1.5 pounds, as in the case of the Louisville engine trial; that the large steam pipe from the Babcock & Wilcox boilers—extending out of doors for hundreds of feet—leaked more or less at the flange joints. But all the conditions enumerated are adverse to the best results by this engine as compared with the pumping-engine. On the other hand, it is urged that this engine runs at nearly twice the piston speed of the Louisville engine. This point has hitherto been considered of much theoretical advantage. I question it, however, especially in view of the many recent tests of slow-speed steam jacketed engines in which the economy seemed really improved by reason of that slow speed. The larger sizes of the cylinders of the Louisville engine should more than compensate for any fancied advantage to the Natick engine, due to its faster reciprocations.

The Natick engine had at cut-off in high-pressure cylinder 20 pounds more steam pressure to its credit than the Louisville engine, and perhaps this is a fair point to raise as a disadvantage put upon the Louisville engine, though I believe that engine would have done no better with the extra 20 pounds than it did do, owing to too small a low-pressure cylinder.

Now for an estimate of the real advantages of either system over the other, as revealed by Mr. Dean's tests.

First, he makes out an apparent advantage in favor of the Louis-

ville engine of 5.3 per cent. I ask, is this figure to be taken as representing the true comparative economies of the two types of compound engine? I believe it is not, and for the following reasons, partly specific and partly general.

At the trial of each engine the M. E. P. referred to the low-pressure cylinder, and the degree of vacuum was: Louisville engine, 24.9 pounds M. E. P., and 13.4 pounds vacuum; Natick engine, 17.46 pounds M. E. P., and 11.9 pounds vacuum. If the load on the Natick engine could have been enough more to have made use of a vacuum of 13.4 pounds instead of only 11.92 pounds, and this decrease in back pressure of 1.5 pounds could have been effected and so added to the M. E. P. of 17.46 pounds, as is entirely possible, and as we should not do on paper, if the proper effect of the better vacuum on the economy of the Natick engine is to be understood, then ($1.5 \div 17.46 = 8.6$ per cent.) 8.6 per cent. more work done by 12.74 pounds of steam would immediately result. The quantity 12 74 pounds is now 108.6 per cent of the amount necessary to do one horse-power of work, so 100 per cent. would be $12.74 \div 108.6 \times 100 = 11.75$ pounds steam as the true comparative economy of the Natick engine, as against 12.16 pounds, that of the Louisville engine, a difference in favor of the Natick engine of 3.5 per cent. I will not try to estimate the harmful effect on the Natick engine duty of poorly piped jackets, insufficient brass tube area in the receiver jacket, or the error in determination of its actual performance due to leakage of steam from the main steam supply-pipe, etc., although it is certainly something, and perhaps considerable. But I should like to point out that there was a leak by the steam valve on the crank end of the high-pressure cylinder, shown pretty clearly by the diagram in Fig. 3, page 4, of the paper on Natick engine.

The point of cut-off shown on this card is (clearance reckoned in) at .19 of the stroke. The point of cut-off shown by the other card is at 22.4 per cent. of the stroke. One would expect to find a lower terminal pressure on the card having the earlier cut-off, instead of which the card showing the fewest expansions gives the lowest terminal pressure. I estimate the rise in pressure due to leakage to have been at least 5 pounds. There appears to have been a loss of steam on the other stroke also, though much less, as I estimate the rise in pressure due to leakage to be as much as 1.5 pounds. Plainly the Natick engine suffers a loss of efficiency by reason of leaky valves, which is not and cannot be correctly estimated. Thus I have shown that, if the effect of all these disadvantages were to be allowed for, the difference in steam consumption in favor of the Natick engine would be materially

larger than 3.5 per cent. I believe I have thereby shown that these data also reveal the engines of the style of the Natick compound as better than ordinary compounds.

Mr. Dean touches upon the theory of the successful operation of the Natick engine in these words: " . . . the economy of the Natick engine is due to high steam pressure with the resultant high degree of expansion, small clearances, and tight pistons and valves." He might have added, "and to the relatively very large port areas," as there is probably no other kind of engine extant having so little clearance space.

Mr. Dean also says, "Although the performance of the engine is remarkably good, the writer believes that it was realized in spite of great defects," but how does Mr. Dean harmonize these apparently conflicting ideas? If this engine does remarkably well, in spite of grave defects, then let us study somewhat the nature of the alleged defects, to find out if such they really are.

To define the Natick engine as simply as possible, it is a triple-expansion engine with the intermediate cylinder omitted, and with an intermediate receiver substituted therefor.

The notion that the only effect of an enlarged intermediate reservoir between the first and third cylinders is to drain water out of the incoming steam and to heat the steam (in case a steam-jacket is used) is one that appears to have taken root in some minds, and I would like now to uproot it. That I may explain clearly what I mean, allow me to refer you to the combined diagram of the Louisville engine on page 8 of Mr. Dean's paper. It may be noted there that no drop occurs at the terminal of the high-pressure card. But what happens on the return stroke? The pressure falls rapidly to a point about in the centre of the back pressure, at least eleven pounds lower than the terminal pressure of the high-pressure diagram. Is this to be classed as "drop" or not? and does it increase the total range in temperature in the high-pressure cylinder? While the bugbear, "drop," is variously defined, still, as it brings with it all the disadvantages of drop, in my view, it is "drop;" it does tend to increase the temperature range in both cylinders.

Now, we read the receiver volume was about seven-eighths of the high-pressure cylinder volume. What would be the effect on the back-pressure line of the high-pressure cylinder diagram if, instead, this volume were, say, three times or more the volume of the first cylinder? Would not the effect be to cause nearly all the "drops" to take place at the terminal of the high-pressure card? It would cause a nearly straight back-pressure line in high-pressure cylinder, at a pressure equal

to the lowest pressure now occurring in the high-pressure cylinder. This would give no greater temperature range in the first cylinder, but it would, on the other hand, considerably reduce the range in the second cylinder. Not a pound of pressure would be sacrificed at cut-off in second cylinder, and the work done by the engine would be slightly increased, although, theoretically, there would be a slight loss of area at the toe of the high-pressure card of the combined diagram.

I ask, would it not be a good thing to do to lower the initial pressure and temperature in the low-pressure cylinder if unaccompanied by any corresponding increase in temperature range in the high-pressure cylinder? But all this would be the result of increasing the size of the intermediate receiver, and it can be obtained in no other way. The mechanical advantage of not striking so heavy a blow on the large low-pressure piston is also considerable, though apart from the phase of the question which I would like to present.

Now, in the test of the Natick engine the receiver pressure was carried relatively higher than I would desire it to be, owing to the fact that it was somewhat underloaded; but still the receiver volume is nearly or quite as large as that of the low-pressure cylinder, and so it has the effect of decreasing uniformly the back pressure on the first cylinder, in this case fourteen pounds. Thus it makes the range in temperature in the large cylinder also much less, and—please mark this statement—thereby contributes to the economy of the engine as a whole. How does it do this? Let this question be answered by a consideration of the grounds upon which the "well-known and fundamental theory of the desirability of equal ranges of temperatures" rests.

This theory asserts that in each of the cylinders of a compound engine an equal amount of cylinder condensation will occur, provided that the range in temperature in each is equal. Could anything be more erroneous on the face of it than that proposition? What account does it take of the fact that the low-pressure cylinder of, say, the Louisville engine has four times the exposed area on its piston and cylinder-head faces that the high-pressure cylinder has? A moment's consideration should show that a unit of area in either cylinder exposed to a degree difference in temperature will, *ot er conditions being identical*, condense an equal amount of steam, unless, indeed, there be some at present unknown dynamic influence upon the incoming steam tending to augment condensation.

Thus it seems to "stand to reason" that in the case of the Natick engine, if the ranges in temperature were maintained equal in each cylinder, with a difference in piston areas of 7 to 1, there

would constantly be many times the condensation occurring in the first cylinder occurring all the time in the second cylinder.

It appears to me plain that the maximum efficiency of the entire engine is reached when there is an equality, not of temperature ranges, but of *amounts of cylinder condensation!* the condensation occurring in the first cylinder being just sufficient to, after re-evaporation at exhaust, take the place of the condensation bound to occur in the succeeding cylinder. Thus, as Dr. Thurston has well said, the most wasteful cylinder in series is the measure of the loss from cylinder condensation; plainly, we can do no better than to make each cylinder equally wasteful by adjusting the range in temperature in each cylinder so as to produce this result.

There is but one way to secure an equality of condensation in the two cylinders of the Natick engine, and that is, as I have just attempted to show, by employing a very large intermediate receiver.

This will of necessity produce some drop at the terminal of the high-pressure card, whereas the intermediate cylinder would prevent it utterly. To the extent that "drop" is a net loss the use of an intermediate cylinder would be a gain, for I realize fully that part of this loss "cannot be recovered by any subsequent event." But after we have admitted this fact we are still no wiser than before; we must arrive at some idea of the net extent of the loss by "drop," that is, the net loss "after all the bills are paid," to use a business man's simile, and then, if we find it to be serious—say, something over two or three per cent.—we can make use of the intermediate cylinder.

Now, it is apparent from an inspection of the combined diagrams (referred to on page 8) that one loss due to making use of an intermediate cylinder is the loss due to wire-drawing in getting the steam out of the first cylinder and into the second. This of itself is a greater loss than the triangular area lost through ordering the point of cut-off at a point on the entire expansion curve that is lower than the terminal pressure in high-pressure cylinder—a fact which is the cause of the drop in the Natick engine. Then a certain, and relatively considerable, portion of the toe of the high-pressure diagram is so much lost work, owing to the fact that it is too little to overcome the friction of the engine, as has been pointed out by Professor Gale.

My belief is that when such practical considerations as those just given are arrayed on the credit side of "drop"—and, be it understood, I here allude to a small degree of drop, say not over 30 pounds—such as we can get along with where the expansions are so many as in the Natick engine, the preponderance of power felt at the piston-rod will be found to be in favor of the two-cylinder rather than the

three-cylinder engine, where pressures of 160, 170, or 180 pounds are to be obtained.

I have dealt with the idea that the office of the receiver in the Natick type of compound engine is to produce drop in pressure at the terminal of the high-pressure cylinder stroke; that there is practically no loss from "drop" in that engine; and that in any compound engine it is necessary to sustain, not an equality of temperature-ranges in the two cylinders, but an equality of condensations. I would now like to look at the question in another light, and will try to show that, leaving the low-pressure cylinder quite out of the account, there is still no greater loss from cylinder condensation in the Natick engine, even though the intermediate cylinder is not employed, than would be the case were it in use.

Suppose the engine to have an intermediate cylinder of a diameter of, say, thirty or thirty-two inches; that is, give the engine what would be a standard intermediate cylinder.

Suppose the three points of cut-off to be so adjusted as to give equal ranges of temperature in each cylinder. We would then have the kind of practice desired by Mr. Dean.

The relative areas of the high and intermediate cylinders are to each other as 1 to 3, and the ranges in temperature are presupposed equal.

Now, it seems to me that, in order to prove that the intermediate cylinder is an "ameliorator" of the loss in the entire engine due to cylinder condensation, it must first be shown that less cylinder condensation, by a considerable amount, gets by the intermediate piston without doing work in that cylinder as steam than would escape from the high-pressure cylinder, were it to be subjected to twice the range in temperature happening when both cylinders are in use, by the instrumentality of an enlarged receiver. Perhaps it is unnecessary to take time to show that the effect of either the intermediate cylinder or of the large receiver upon the conditions under which the low-pressure cylinder takes its steam is identical in either case, so that, as I have said, that cylinder may be left out of account, in calculating the deleterious effects on the economy of the engine by reason of leaving out the intermediate cylinder.

The question, therefore, is: "Does more condensation and re-evaporation take place in the high-pressure cylinder—having twice the temperature-range and one-third the area of the intermediate cylinder—than takes place in the intermediate cylinder, if used?"

To ask this question is also to answer it, I think, in the negative, in the light of what has been said above.

To return to the author's indictment, that the Natick engine labors under great defects; I have mentioned that many of the defects, such as defective jacket circulation and defects of that order, are of themselves a sufficient cause of the difference in economy actually observed between the two engines; I agree fully with him in the abstract proposition that the highest economy to be realized in the perfect engine—that is, in one having non-condensing cylinder surfaces and frictionless parts—is to result from the combined influence of two conditions—using a volume of steam at the highest possible pressure, expanded the utmost number of times.

The Natick engine is the embodiment of this principle, so far as the principle can be embodied. It uses steam of a higher pressure than does any other compound mill engine of which I have any knowledge.

It expands a volume of it sixty per cent. more times than the ratio of expansion in the Leavitt engine—the ratio being 1 to 33 for the Natick engine and 1 to 21 for the Leavitt engine. If the Leavitt low-pressure cylinder had been fifty per cent. larger it would have enabled the expansions to be on a par with the ratio in the Natick engine; but the increase in economy would only result, I venture to predict, if an enlarged receiver were also used.

But as to the size of the receiver and the volume-ratio in the Natick engine being great defects in its design, I confess, for reasons stated, I cannot see it quite yet in that light, and mistrust I shall never be able to see it so, unless I am given more information of a kind contrary to that now in my possession.

There is one other minor and *last* aspect of the matter that I might bring briefly to your attention by quoting the trite saying: "One swallow does not make a summer." This Louisville engine has not only broken all previous records, it has left them out of sight; they are not even in the race at all.

Note the performance of the Pawtucket compound pumping engine; note that of the great Allis tandem compound at the Plymouth Cordage Works—fifteen or twenty per cent. less economical, though under fully as good conditions. Note that of the triple-expansion Laketon pumping engine, working with steam at one hundred and fifty pounds, and yet fifteen per cent. less economical. Note that of the European triple-expansion mill engine, the Sulzer Corliss, of large size and splendid design, yet out distanced by this Louisville compound six per cent. In such company I confess I believe the performance of the Natick engine—improved upon by itself, as it doubtless could be, several per cent.—is not unsatisfactory enough to warrant an impeach-

ment of its design, especially when four other engines of the same type have all given equally good or better accounts of themselves; whereas we cannot, with certainty, get a plain compound Corliss *mill* engine to do as well as fourteen pounds, try as we will.

Dr. Emery.—It is known by many present that several of the problems under discussion were examined by me about twenty years ago. The lessons then learned have not lost their force in many respects. The later engineers have had an opportunity of experimenting with higher steam pressure and more perfect mechanism, and have obtained much more economical results; but it is a question if such results are not due entirely to these two features. I class reduced clearance with more perfect mechanism, for the reason that the mechanical details of the engines were substantially the same then as now. There is a tendency, however, to theorize, as to features other than those mentioned, and we are fast reaching a condition of ultra theory and ultra expansion, like that developed for the older type of engines during the war, when the *Winooski* and *Algonquin* ran their celebrated dock race here in the city of New York. It will be recollected that on the last-named vessel 15 to 20 expansions were attempted in a single cylinder with 80 pounds steam pressure, while, in the other vessel, designed by Mr. Isherwood, 45 pounds steam pressure was used, cut-off at about $\frac{6}{10}$ of the stroke, but with a valve moving so slowly that the virtual cut-off was at about $\frac{4}{10}$. The low-pressure steam machinery pulled more steadily than the other, used *less* steam per horse-power, and did not break down, whereas that using the high-pressure did. This showed that there was more to the question than mere theory. In one case the expansion was carried to an extreme unwarranted by the conditions, so that the more simple machinery, with less expansion than was warranted, gave the better results. History repeats itself; and very similar results are coming to light in relation to triple compound engines compared with compound engines, which show it is time to call a halt and ascertain what points have been actually settled by previous practice. In discussing Mr. Webber's paper I called attention to the very low mean pressure in the large cylinder, and made the statement that the work done in the intermediate cylinder could have been transferred to the larger cylinder, and greater economy thereby secured in that cylinder. It follows that, even if the gain in the low-pressure cylinder was balanced by a corresponding loss in the high-pressure cylinder, the economy of the simpler compound engine would still be as great as that of the triple compound. I was very much surprised to hear the statement, in Mr. Rockwood's discussion of the present paper, that he would have preferred to have the engine which he

designed operate with as low a mean pressure in the low-pressure cylinder as I had criticised. We ought by this time to know all about the results with low-pressure steam, as very many experiments have been made with it. Mr. Isherwood's books are full of such experiments. Those made on the *Michigan*, at Erie, Pa., settled many of the questions, though others are equally applicable, more particularly those with which the speaker was connected, known as the "Novelty Iron Works Experiments," of which a table has been published, without explanations, in *Appleton's Cyclopædia of Mechanics*, and Vol. II, American edition, of *Weisbach's Mechanics*. The general result is well-known. Engines using 15 pounds of steam were more economical than those using 5 to 10 pounds; 25 pounds pressure was found still more economical, and 40 pounds more economical yet. The last-named pressure is at present out of the question for the large cylinder of a compound engine. In fact, there would be some gain by compounding with such pressure, but in regard to using steam at a pressure below that of the atmosphere, and at 10 or 15 pounds above it, there is no question whatever; the latter is much more economical. The terminal pressure in a low-pressure cylinder should be high enough to insure thorough drainage by the sudden expansion during the exhaust, the gain in this way being greater than the loss caused by reducing the expansion in such cylinder. In the design of modern compound and triple compound engines we should start with the maximum already obtainable with a low-pressure cylinder; that is, do as much work therein as has proved economical in low-pressure practice, then obtain as much work with the steam above that pressure as is practicable. The result will be that more work will be done in the low-pressure cylinder than in the high-pressure, as is, indeed, shown by the tests of the Leavitt engine, now under discussion. This does no harm. We have simply to provide for it in the design, even if two low-pressure cylinders are used, as in some forms of compound engine.

I wish to thank Mr. Rockwood for the very earnest work he has done in developing this question of compound *vs.* triple compound engines, though I do not think he is right in making such an extremely large ratio of capacity between the high and low-pressure cylinders. It is also a source of gratification that even better economical results have been obtained with a Leavitt compound engine, and as the latter result was secured with a less number of expansions, and with a larger proportion of the work done in the large cylinder, it indicates the correctness of the principles above stated.

The general conclusion appears to be that we cannot as yet carry the steam pressure high enough to make the triple compound engine of

value in a commercial sense. It is true that the best triple compound engines have given a little better results than the best compound engines, but fairly large percentages of gain are for such economical engines very small quantities, and are easily wiped out by very trifling derangements, such as small leaks, want of care with jackets, etc., and are readily balanced by other items of cost, such, for instance, as a little higher wages of the engineer or the greater interest due to increased first cost. The coal is only one of the several items of cost in operating a steam-engine, and all must be considered in making a commercial balance.

In making these remarks I wish to encourage rather than hinder any attempts to obtain the very best results possible. The chairman will realize that the anthracite supply is limited and that that kind of coal will appreciate in price, so the very extreme economies will be valuable in the future, even if not warranted by commercial considerations in the present.

This discussion of compound *vs.* triple compound engine will be more valuable than seems at first sight. I have already called the attention of the American Society of Naval Engineers to the subject, with a view of saving the weight and to some extent the space occupied by the intermediate cylinder on board ship. There the elements of space and displacement are of the greatest importance, and, moreover, the full power runs are comparatively so short that some economy can be sacrificed under such circumstances, if economical results are obtainable at ordinary cruising speeds. It is true that the three-crank system of the triple compound engine is a desirable feature in producing smoothness of working, but it need not be sacrificed if a return be made to compound engines, as two of the cylinders can be low-pressure cylinders, as was, indeed, a common practice when lower steam pressures were employed. The system of doing more work in the larger clyinders, previously recommended, aids in the solution of this problem, though doubtless there will be some difficulty in distributing the load equally to the three cylinders. The system adapts itself very well to the conditions of varying loads obtaining on board ship, and I have no doubt that in due time valuable developments will be made in this direction.

Mr. William Kent.—Mr. President, I think that in years to come engineers will read with great pleasure this paper of Mr. Dean's and the discussion by Mr. Rockwood, supplemented by Mr. Emery's discussion. As the matter stands now we can say we have learned that the Leavitt engine, according to Mr. Dean, is about five per cent. superior in economy to the Natick engine, and according to Mr.

Rockwood, if we make the proper allowances, the Natick engine is eight per cent. better than the Louisville engine. Add these figures together and divide them by two and we have the two engines very nearly alike. Mr. Rockwood mentioned in his remarks the Pawtucket engine, and it was also in my own mind at the time, as to what is the cause of the difference in economy between the Louisville engine and the Pawtucket engine. The Pawtucket engine had 16 expansions as against 20 in the Louisville and 33 in the Rockwood engine. The Pawtucket engine had 120 pounds of steam pressure as against 151 in the Louisville and 175 in the Natick engine. The Louisville engine had high vacuum as compared with the Pawtucket. The Pawtucket engine had only 240 for piston speed as compared with 371 for the Louisville and 611 for the Natick. It is probable that if the Pawtucket engine had been given 150 or 175 pounds pressure of steam and if the expansions had been 20 or 33 instead of 16, it would have shown a better result. So that the Pawtucket engine might stand pretty near the top if you would only give it the advantage these other engines had in steam pressure and expansion and piston speed. We cannot make a satisfactory comparison between the Natick and the Louisville engines, because the conditions are so different. The Louisville engine had 20 expansions. Was that the best practice for that particular engine? The Natick engine had 33. Was that the best expansion for that particular engine? The steam pressure of the Natick engine was 175 against 150 in the Louisville engine. Were these pressures the best for those particular engines? Of course, the vacuum in the Louisville engine was best, at all events, and the Natick engine would, no doubt, have been glad to get such a vacuum. But I say we cannot satisfactorily determine which is the best form of these two engines until they are both tested with the same steam pressure and vacuum, and until each engine is tested with varying expansions, until they find the expansion best suited for that particular engine.

In regard to Mr. Emery's remarks, he makes a point about the compound *vs.* the triple-expansion engine for marine practice It is strange that, about 1882, the very engine he speaks of, the compound engine with three cylinders, was the favorite engine, and it has paid since that to take these out of the ships and substitute the triple-expansion at great cost, a great economy resulting from the change, although I admit that putting in boilers of higher steam pressure might have been largely the cause of the economy. We cannot determine that, however, because we have not had a trial of that particular form of compound engines with high pressure and with moderate pressure steam. We do not know to-day what that engine might have been

capable of doing with steam of one hundred and seventy-five pounds pressure, because it never was tried.

Mr. R. S. Hale.—I should like to ask Mr. Dean what was the slip of the engine, as determined by weir measurement. Last summer Mr. Brackett spoke to me of something like seven per cent., and if it was as much as that, would it not change considerably the friction of the engine and the duties, as figured, of the plunger displacement?

Mr. Platt.—I would like to ask for some information with reference to the boiler practice at this mill engine test. The contrast between the two tests of five hours and one hundred and forty-four hours, without any explanation with reference to the boiler practice, seems to me to leave something lacking.

Mr. Dean.—The tests of these engines as they are reported were simply a feed-water test, and the water in the boilers was at the same height at the end of the test as at the beginning, and the steam pressure was the same. That, with a little experience, can always be brought about.

That part of the test of the Louisville engine referred to by Mr. Hale was not touched upon by me in this paper, for the reason that I felt more interested in the steam performance than in any other part, and thought probably other people would also. The slip of the Louisville engine is remarkably large.. It was so large that it took us some two or three days to try to find out the reason, and it averaged about 6.8 per cent. We determined the data for every day. In fact, we made six different consecutive tests; that is to say, as I just stated, we determined the data each day separately, and it came always about 6.8 slip. We were somewhat suspicious of our weir. But all of our suspicions, so far as we were able to see, proved to be groundless. We several times stopped the engine, and shut valves in the main, and noted the flow of the water in the chamber at the weir, and we also did that when the valves were not shut, so that it all came on the pump-valves. At such times the flow of water was about one per cent. of the amount of the plunger displacement. The only way that we can account for this unusual slip is this—that the pump valves were metallic valves seating on metallic seats, and the Ohio River water carries considerable sand in it, and those valves in a short time scored themselves out more or less, and valves that were taken out seemed to be gouged out as it were on one side, and not all the way around. But, of course, we can hardly imagine a pump-valve seating squarely. There is a spring, as a general thing, to press it down, and that spring will probably carry down one side a little quicker than does the other, and of course all pump-valves must be loosely fitted so that they will be free to move

under all conditions. It looked as if the valve in general struck on the edge and gouged the seat out, so that we thought that probably a good deal of the slip was to be accounted for in that way; and in listening to the pumps, putting one's ear right against the pump chambers, there was a sound which did indicate that there was water going through somewhere—it was rather difficult to tell where—at the time when the water was being forced up into the main. But I do not know that I can throw any additional light on that subject. The whole matter was an astonishment to all of us, and we used a good deal of time to try to find out what the trouble was. That, however, would not, as Mr. Hale suggests, affect the friction of the engine.

Now that I am on my feet I will speak of some other interesting things which were done with that engine, and which are not mentioned in the paper. The result was so unusual that I thought I wou'd go to rather unusual pains to corroborate it, and in the report—I have forgotten whether I stated in this paper or not—but in the report of the test it is mentioned that the condensation in the jacket was determined by passing it through a Worthington meter, which meter worked with remarkable steadiness. It always showed about twenty-five cubic units per hour; whether you took the data on the first day, or third day, or last day, it was just the same. Immediately after the trial I calibrated that meter for some three hours. I was hardly content, however, with that calibration, and after I got home I wrote to the chief engineer of the waterworks to ask him to determine that condensation for me by actually weighing the jacket condensation, and also to run another test of twenty-four hours' duration; and I will say here that Mr. Hermany had a very competent chief assistant, who helped me in this test and in whom I had the utmost confidence. He fully appreciated the necessities of the case. Persons who have read the account of the test in the report will remember that the amount of water by the feed-pump was determined by computing it from the rise of temperature of the water before it was heated by this pump exhaust, and after. But in this supplementary test which I asked Mr. Hermany to make, the exhaust was turned out of doors and the feed-pump was run by the donkey boiler, and the jacket water was actually weighed throughout the twenty-four hours, and the separator condensation also. This jacket condensation differed from that which I had determined by .06 of one per cent. The head of water on the pump was almost identical, the revolutions were just the same, and the indicated horse-power figured out precisely the same as on the official test. On each of the six days of the test the amount of feed-water used by the engine was 187,000 pounds, almost without exception. It differed only a small number of pounds.

The greatest difference that we found from my results was the separator condensation for the twenty-four hours. I made it 3,900 pounds in twenty-four hours, and he made it 2,800. There was a difference, you see, somewhere about one-half of one per cent. of the total feed. We are dealing with such large quantities that it is of no importance. He also ascertained for me the two jacket condensations separately, and the re-heater condensation separately, but simultaneously. All of the data which are given in my report have been so thoroughly corroborated and reproduced day after day on that test that they are singularly to be relied upon.

*Replying to Mr. Ball.—As to the effect of compression on economy, the experiments to which he refers as having been carried out by Professor Jacobus were made on a relatively low-grade engine. By that I mean a single-valve engine with large clearances. Results from such an engine, I believe, are little or no guide in determining practice with high-grade engines. By high-grade engines I mean four-valve engines with small clearances. With low-grade engines some thermo-dynamic phenomenon with high compression may creep in which overpowers others. In the high-grade engine there is less room for erratic phenomena, and we can work more closely to our theories and obtain corresponding results. The Leavitt engine is worked out in detail close to the theories, and the results are given in my paper.

Mr. Ball's arguments do not appeal to me, either with reference to compression or to drop.

With reference to drop, I will simply say that the modern engine is made to use steam expansively. It may be done in one cylinder, but it has been found that it is much more economical to divide it up into steps, each cylinder performing a step. Why should not one step begin where the preceding one leaves off? I confess that I never have been able to see.

As I understand it, Mr. Ball claims advantage in drop, because it superheats the steam. If we assume steam of 45 pounds absolute to drop to 25 absolute, and thus to drop 20 pounds, the superheat will be $\frac{1165.6-1155.1}{0.48}=21.87^\circ$. This superheat would not, however, exist, for the released heat would find itself in wet steam, and therefore the supposed benefit is all but *nil*.

The amount of heat added to a pound of steam of the lower pressure would be $1165.6-1155.1=10.5$ B. T. U., or $\frac{9}{10}$ of 1 per cent., and this, in turn, would dry out $10.5 \div 922=0.013$. or $1\frac{13}{100}$ per

* Author's closure, under the Rules.

cent. of moisture in the steam, the benefit of which is unknown, but small. In order to secure this small benefit Mr. Ball would lose expansive energy of the steam the value of which is exactly known, and is represented by $1 + \text{hyp. log } \frac{45}{25} = 1 + \text{hyp. log. } 1.8 = 1.5878$ per pound of steam. I prefer to get this work out of the steam, especially when its quality is restored by a re-heater.

Replying to Mr. Rockwood, I think it is not unreasonable to suppose that the whole engine at Natick was built in accordance with the Rockwood system, and therefore to be criticised as such. I am somewhat in sympathy with Mr. Ball in not understanding what the Rockwood system is. If it is a ratio of 7 to 1 it is seldom made by him, and has in a general way been put into English steamships several years since. As for tests of triple engines with the intermediate cylinder cut out, the columns of *Engineering* contain the results of tests that are disastrous to the cut-out.

Mr. Rockwood states that the steam-pipe leaked at the joints at Natick. This is not true, except at a slip expansion joint, and the amount was so slight as to make no perceptible difference in the result.

The shortness of the tests of the Natick engine are unfavorable to it compared with the Louisville.

I agree with Mr. Rockwood that, so far as I know, the advantage of high piston speed is mythical, and cuts but little figure in the comparison made by me.

The probable greater economy that would be due to a better vacuum with the Natick engine was estimated by me by ascertaining how much more area would have been given to the L. P. indicator diagram thereby, and the resultant increase in work done by the steam.

Mr. Rockwood entirely misinterprets the sag in the exhaust line of the H. P. indicator diagram of the Leavitt engine, except that it increases the range of temperature in that cylinder. He, however, makes no allowance for the fact that this increase in range is corrected by the rise in pressure in the receiver and H. P. cylinder after the closure of the L. P. inlet valve. This correction is intentional, both to avoid drop in pressure and net drop in temperature.

To make myself understood with reference to drop I will define it. In general, drop is a fall in pressure between the end of expansion in one cylinder and the beginning of expansion in the next, and, specifically, in a tandem or Leavitt engine, it is the fall in pressure between the terminal pressure in one cylinder and the initial pressure in the next. The only unavoidable drop in such engines is due to the work of moving steam from one cylinder to the other. In the Leavitt engine, expansion in the second cylinder begins with the beginning of

the movement of the piston from the end of the stroke, and no fall in pressure can occur until piston movement begins; while drop is unresisted expansion, or fall in pressure without piston movement and without doing work, and is therefore a dead loss, except in so far as the heat released produces some superheating. It must not be overlooked that true expansion takes place with the whole stroke of the Leavitt L. P. piston, only that the law changes after L. P. cut-off.

Mr. Rockwood is wholly wrong in supposing that a larger receiver would produce drop in the release end of the high-pressure diagram. This has nothing whatever to do with it, as a drop, or its absence, will be determined by the low-pressure point of cut-off in either a tandem, Leavitt, or cross-compound engine. If valves are properly set in either of these types of engine, and the cut-offs of all but the first cylinder are not affected by the governor, and a permanent *regime* has been established, neither will ever produce drop or loops in the indicator diagrams, except the always unavoidable drop above mentioned, and which increases with speed. This is furthermore entirely independent of the receiver volume, or point of cut-off in the high-pressure cylinder. The large receiver will diminish the temperature-range in the high-pressure diagram, and is so far beneficial unless the correction above referred to is wholly effective. It will, however, not affect the range of temperature in the low-pressure cylinder, as Mr. Rockwood claims, because the initial and back pressures in this cylinder are not affected by the receiver.

My understanding of the effect of equal ranges of temperature in cylinders is not, as Mr. Rockwood says, that "an equal amount of cylinder condensation will occur," but that a minimum total condensation will occur. Although I cannot now give an absolute proof of this, I am satisfied to hold this view for the present. The theory that Mr. Rockwook tentatiously advances, viz., that equal range takes no account of the amount of cylinder surface, and that the large cylinder would necessarily condense much more than the small, is inconsistent with facts, for we know that in every engine the condensation is greatest in the small cylinder.

Finally, after all has been said and written, the fact remains that an engine with a cylinder ratio of 4 to 1 has surpassed in economy an engine with 7 to 1, carrying a higher steam pressure.